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SCIENCE, THE NAVY AND THE FUTURE*.

BY CAPTAIN ROBERT D. CONRAD, U. S. NAVY, (RET.).

In discussing the Navy's position in science, I must begin with a brief sketch of the principal parts of the naval organization which deal with scientific and technical matters. A clear understanding of their responsibilities and interrelationships is essential to an appreciation of how we function.

The most significant recent addition to the Navy, from this standpoint, is the Office of Naval Research. It was created by the Secretary of the Navy about two years ago, and was originally called the Office of Research and Inventions. Congress passed

* This article is taken from an address given to the Industrial Research Institute at a meeting in Chicago on February 28, 1947. Captain Conrad was in the office of Naval Research at that time. He is now assistant director of the Brookhaven National Laboratory at Upton, Long Island, N. Y.

a law in August 1946, which gave it statutory authority and its present name. It functions as a main office in the Navy Department for initiating, conducting and coordinating fundamental research, along with other duties specified by law.

There are four main divisions of the Office of Naval Research: the Naval Research Laboratory near Washington, the Special Devices Center at Sands Point, Long Island, and the Patents and Planning Divisions in Washington. Branch offices have been established in Boston, New York, Chicago, San Francisco, Los Angeles and in London.

Another important addition to the Navy is the Operational Development Force, a seagoing task force under the Chief of Naval Operations. It is a highly flexible organization of ships, submarines and aircraft of all types, whose mission is to test and appraise all new weapons and devices under actual service conditions. It both reflects and discloses the requirements of the Fleet from the critical and independent viewpoint of the user.

The great bulk of technical responsibility in the Navy rests, as it always has, with the material Bureaus. The relationships of the Bureaus with industry, directly and through the naval shipyards and naval laboratories, have made the Navy the tremendous technical organization that it is today.

Thus, in broad outline, three agencies are involved in keeping the Navy in harmony with the progress of science. It is the business of the Office of Naval Research to foster the creation and evaluation of knowledge; the Bureaus collectively translate knowledge into things; and the Operational Development Force applies the acid test. In the simplest terms, these functions are research, development and evaluation. The three functions are inextricably intermingled, as the director of any laboratory well knows. Each one reacts upon and is influenced by the other two. Yet they represent three quite different points of view, no matter how shadowy and indistinct their boundaries; and the three responsible naval agencies I have described are therefore independently managed, under the general policies and fiscal controls of the Navy Department itself.

Now this general plan is so important that it is well worth our time to examine it more closely. The Office of Naval Research

may initiate any research project it considers desirable, either in its own laboratories or by contract; but it must collaborate with the Bureaus in the application of research results on the one hand, and with the operating forces in exposing fresh research problems on the other hand. The Bureaus may undertake the development of such weapons and equipment as they see fit, but must also be guided by the possibilities revealed by research and by the requirements and limitations of operational utilization. The operating forces may specify their needs, but their planning must be in line with the latest advances and possibilities of science and technology as interpreted by the Office of Naval Research and the Bureaus. It is an extremely effective interlocking arrangement.

The analogy with industry is fairly obvious. The Office of Naval Research corresponds to the central laboratory of the firm, under its own vice-president. The Bureaus correspond to the manufacturing plant, with its production shops, engineering departments and works laboratories. The Operational Development Force corresponds to the evaluation and sales departments. We believe that it is sound business practice for the Navy, as well as for industry, to separate the management of these functions; and that subordination of any one to another would weaken the effectiveness of all.

Since management questions are one of the main concerns of the Industrial Research Institute, you will perhaps be interested in some of the management aspects of the Office of Naval Research. I have selected the research contracting part of our business for discussion, as it constitutes not only a new departure for the Navy, but is being regarded as a significant advance in the relations between civilian scientists and the Federal Government. First, however, it is necessary to define our terms. Research has acquired such qualifying adjectives as "basic", "fundamental" and "applied". Other terms, including "science", "engineering", "technology", "development" and "design" have a variety of usage which invites confusion. The two I shall use most are: *Research,* meaning the *search* for new knowledge of nature, and *Development,* meaning the *application* of knowledge. In accordance with these definitions, research is not objective, whereas development aims at specific results. Research predominates in universities, while development is that main business of an industrial laboratory. Each is closely associated with the other, and they can seldom be entirely separated except in principle.

It is now generally appreciated that our wartime scientific activity consisted of developing the pre-war stock of knowledge accumulated by research into the weapons and devices which the war demanded. Very little research was done during the war; there simply was no time for it. Toward the end, it began to appear that all we could hope for was the continued modification and improvement of the things we already had. The stream of genuinely new ideas had pretty well dried up. And to make a bad matter worse, a shortsighted draft policy had taken many future scientists from their studies, so that the new crop was blighted. No one deplores the shortages of scientific personnel more than you do, I am sure, for it is probably the greatest single barrier to economic recovery and continued progress.

Still other difficulties that emerged as the war drew to a close concerned the situation in universities. Sources of income were shrinking, staffs were scattered, research was becoming more expensive, and crushing overloads of teaching were inevitable as the G.I. Bill of Rights brought unprecedented numbers of students to the campuses. The overall prospects for reestablishing strong and active research programs in the country were dim indeed. Now all this was a matter of grave concern to the Navy. With the realization that research must be supported, that it provides the foundation for all new developments, and that there was no civilian agency in the Government in a position to provide assistance, the newly-created research office in the Navy began functioning in May of 1945.

The difficulties we have had in building up the Office of Naval Research during a time when all other activities of the Navy were shrinking, and in developing effective procedures without any precedents to guide us, can be better imagined than described. I am happy to report that we have succeeded beyond our expectations, and that we have not only found our place in the Naval establishment, but have received firm and widespread support from the civilian scientific fraternity as a whole. At the present time*, we have placed approximately 350 projects covered by nearly two hundred contracts, which total upwards of twenty-five million dollars. These projects are distributed throughout the

* February, 1947.

various fields of physical and medical sciences. About three-quarters of them are placed with educational institutions, and the remainder with industrial laboratories. Thus these funds are helping to alleviate the shortage of scientists, since many students are employed on them.

We have consistently adhered to three principles in building up this enterprise.

First: we do not attempt to solicit research proposals. We have circulated explanations of our general philosophy of operation, and have announced that proposals will be welcome; but no one has been urged against his better judgment to submit a proposal to us.

Second: we make no attempt to direct the work once the contract is signed. The broad objectives are initially formulated by mutual agreement; but from that point onward, the contractor has complete freedom in handling his work. Research is subjected to such intense criticism among scientists themselves that unnecessary duplication—which would be the only excuse for an attempt to direct it—is avoided automatically.

Third: most of our projects are unclassified. Secrecy stifles progress, and unless papers are published and discussed, true research simply cannot flourish. We expect a scientist to make confidential or secret reports to us if he wishes to point out the national security potentials of his work, and to limit his published material to its scientific aspects only. If the desirability of developing some weapon or instrument for naval use is indicated, the research group is given the option of accepting a classified status for the job or having it removed to another contractor or to a government laboratory.

So much for our general policies and ground rules. I come now to a description of how we are actually organized and how we handle our work. And bear in mind that I shall deal only with our contract business; time does not permit a discussion of the affairs of our two main research establishments—the Naval Research Laboratory and the Special Devices Center—nor of our work in patents, or of the operation of our field offices. Research contracts are mostly handled by the Planning Division, and my discussion applies mainly to it.

The Planning Division of the Office of Naval Research is

organized on a functional basis. It contains four branches: one for the physical sciences, one for the medical sciences, a program branch and one for administrative matters. Each of the scientific branches is comprised of a number of sections, which are identified by the names of the fields of science in which they are primarily interested—mathematics, geophysics, physiology, biochemistry, physics, and so forth. These sections in the scientific branches are staffed mostly by civilian experts. The function of the program branch is to insure that the results of research are correctly interpreted in terms of the Bureau development programs, and to exercise a coordinating influence upon all research and development in the Navy Department. It also consists of various sections, named for the principal development areas of current interest; and it is staffed mainly by naval officers. Each of the sections is alloted a share of the budget; and within their respective fields, the Section Heads have a good deal of autonomy.*

Civilian scientists and officers have equivalent responsibilities and delegations of authority. Two conferences are scheduled each week, one to deal with the managerial aspects of our business, and the other a symposium conducted by the various Section and Branch Heads in rotation. Special meetings are held jointly with civilian scientific groups, and representatives of the office attend most of the meetings of the learned Societies. I do not believe that there is any group more actively in touch with research on all the frontiers of science than this one in the Office of Naval Research.

A research proposal usually reaches us as an informal letter, and is routed to the scientific section most directly concerned. Here it is carefully scrutinized from several points of view:

(a) the known competence of those who propose it
(b) the extent of our investment in the field
(c) the potential importance to the Navy
(d) the cross-relationships with other fields of science, and
(e) the extent of government support already furnished the institution concerned.

The reasons for this screening are quite obvious. Our limited funds must be invested as wisely as possible, and we wish to

* The organization has changed somewhat in structure since this was written, but it is functionally the same.

avoid a preponderance of government-supported research work in any establishment, particularly educational institutions. These various considerations have resulted in the rejection of about half the proposals which have been submitted to us.

The Section Head reports his decision to the Branch Head, and if it is favorable, the section then prepares a procurement justification. This serves as a notice to the contract section that negoitations between the business people should be undertaken. The product of the negotiations is a procurement directive, which contains a complete record of the negotiations and of the decisions reached; and if no unforseen difficulties are encountered, the actual contract is drawn up and signed shortly thereafter.

It is customary to speak of planned research programs these days, but this is almost a contradiction in terms. Just as an unexplored country cannot be mapped, so it is futile to plan what research shall accomplish, or to direct it toward a preconceived objective. Our program in any given field is simply the sum of our projects in that field, as submitted by scientists and approved for contract, plus the projects in our own laboratories. Aggregates of related projects have emerged, with a good deal of spontaneity, as recognizable programs, and have been crystallized as such by discussions and conferences.

This is, we believe, a sound procedure, although it may appear haphazard at first sight; for it represents, in the final analysis, the judgement of scientists on what they consider most important. No profession is more self-critical and exacting than scientific research. Each scientist is dependent upon the knowledge he inherits from his predecessors and the reliability of the work of his colleagues. Thus the aggregate of projects is forced by the internal discipline of science into a consistent pattern. It is an amazing experience to witness the growth and the interlinkages of such a dynamic structure.

The administrative procedures for regulating so diversified a business are necessarily complex. I shall mention only a few of them. A good deal of travelling is required. Each member of the staff must submit a brief report of each trip he makes. These are filed by subjects and by places visited, and the files must be consulted by anyone planning a trip in order to prevent useless travel and unintentional interference. Budget allocations are

continually watched, and balances are struck at least once a month. Correspondence standards need constant monitoring. Personnel and space problems are always acute and change with bewildering rapidity. All of these administrative matters, and many more, must be handled in relation to the overall administrative staff which is responsible for the affairs of all of the components of the Office of Naval Research.

The system we have found most workable, after considerable trial and error, is a dual line of authority: one for all scientific, technical and policy matters, and the other for administrative services and controls. Thus each section has one or more administrative people in it, who are under the supervision of the Administrative Officer. All details of office routine are thus removed from the shoulders of the branch and section heads. The scientific and technical personnel are under the general supervision of the Chief Scientist, who is responsible for the scientific standards of the work and the wise choice of our investments. The resulting organization is completely flexible, and permits the free interchange of views so essential to a balanced and progressive endeavor.

A large share of our time is devoted to interdepartmental and interagency work. There are a good many committees in Washington, as you all can testify; and we attempt to keep in touch with each of them which in any way concerns research and development. We supply members to as many as we can. We are also beginning to exchange personnel on a rotating basis, and the staff now includes Army, Marine Corps and Coast Guard officers.

Keeping records and furnishing information is, of course, one of the main problems of a headquarters organization such as ours. An up-to-date library must be maintained, containing periodicals, contractor's reports and government documents as well as books. Summary records of all research and development work of the Navy, in all its Bureaus and Offices, are kept posted, plus lists of the similar work in other agencies. A technical information service is supplied for internal use. Literature surveys must be provided beyond the scope of our own collections. All of this has proved too much for our limited personnel, and we have recently enlisted the expert assistance of the fine staff of the Library of

Congress. We also have a small editorial group who carefully reworks all material for our reports and publications, for we have learned a sad truth known to all research managers—that scientists do not always include good writing ability among their talents.

I turn now from the affairs of the Office of Naval Research to more general and even prophetic matters. I would like to point out, for instance, that industry has a magnificent opportunity to enlarge its contributions to research. Admittedly, an industrial laboratory's main job is a development one, designed to serve a particular industry. But the spark that gives life to a laboratory is the real research that is done, no matter how small it may be in relation to the whole. The wise laboratory director opens opportunities for research, and the last thing he does is try to direct it. He is rewarded by an improved morale in his organization, and by unexpected discoveries which may increase the profits of the parent company, and which at least will raise its prestige. Just how far he can go in this direction lies between him and the board of directors; but a laboratory without some bit of true research on its books will become a pretty dull place.

Nor is this all. Industry employs many scientists; and, by this token, industry has a reciprocal responsibility to science. Industry as well as universities can train scientists, inspire young men and women, and offer educational opportunities which can scarcely fail, however indirectly, to benefit the industry itself. The educational programs of industry, in many cases closely connected with those of educational institutions, are generally excellent and should be enlarged.

Research has traditionally been a slow and often exasperating process. In universities especially, the scientist has little help. He must frequently be his own machinist, glassblower and technician. Most of his time is spent on irksome tasks for which he is not particularly well qualified. It is a shocking waste of his time and talents, for once he knows what he wants done, much of his work can be performed better and more rapidly by others.

Conditions are better in industrial laboratories. Here the research worker not only is supplied with many laboratory services that are conspicuous by their absence in universities, but he may also draw upon the resources of the parent company for all sorts of special skills and techniques to aid him in his work. I would

like, therefore, to urge the formation of close working relationships between industrial and university scientists, and especially that scientists seek the aid of industry wherever possible. Just how this is to be done is a local problem, and situations vary so widely that any further general remarks become platitudes. There are, I know, many instances of fruitful associations of scientists in both camps. But the possibilities of mutual benefit by pooling talents and facilities have really been hardly scratches. This is very noticeable in medical research, where progress could be immensely accelerated if the complex, tedious and painstaking studies in biochemistry and biophysics received more attention from industrial research laboratories.

Returning to the question of national security, it is clear that the nature of war has irrevocably changed. Never again can we view war as a duel between armies and navies and air forces, shielding the civilian population from its terrors. As put by my friend Luis deFlorez, the earth has become so constricted by improvements in transport and communications, and the destructive power of weapons has so increased, that there will be no room for sideline spectators in any future conflict. Entire populations will be involved. Industrial areas will be primary targets. The keystone of our strength now lies in our scientific manpower, and on our abilities to create knowledge and transform it into the means for self-preservation. Our safety lies, to a very great extent, in the maintenance of relationships between scientists and the Armed Services. The industrial research laboratories play an extremely important part in these relationships. I would urge that each industrial laboratory devote at least a small portion of its time and talents to problems of national defense, in collaboration with the Armed Services. Let us not forget that we shall not again have a period of grace for getting ready if war should come again.

I have talked a lot about national security and the threat of war. These are the great issues in the world today, and concern for them underlies the thoughts and actions of everyone everywhere. Doubts, fears and uncertainties disturb our minds. Sometimes it seems that our achievements have outrun our intelligence, and that our affairs have become so tangled that we can no longer manage them. Yet I think that all these difficulties are evidence of a resurgence of the mind and spirit of mankind. Two

months ago I prepared an address for the Commonwealth Club of California which compared the state of things today with the period known as the Renaissance in Western Europe, five centuries ago. There is much reason to believe that a new Renaissance is approaching today, and I shall endeavor to indicate a few of the signs that herald it. These signs are not to be found in material things. Progress in power, transportation, communications, and even in the arts, does not constitute a renaissance, but only a somewhat unreliable manifestation of it. The precursors must be sought in intensifications of intellectual activity and in new attitudes of mind. I think they are especially visible in science, but a few examples are all that time permits.

Since the turn of the century, a completely new way of thinking about physics has emerged. This is built around two ideas: relativity and quantum mechanics. Relativity has brought time and space together and given a new concept of the universe. Quantum mechanics has synthesized matter and radiation, and brought fresh understanding of the ultimate particles of Nature and the forces between them. These ideas together are nothing short of cosmic in their grandeur. We accept without question the notion that the earth revolves about the sun, and that the solar system is but an infinitesimal part of the heavens. Yet this was such rank heresy when Copernicus announced that it was one of the main causes of the Inquisition. How long it took for people to believe it! Think what repercussion it produced to realize that our earth was not the center of the sky, nor the biggest object among the stars. Religious beliefs had to be overhauled, and the whole philosophy of man's place in the scheme of things required a fresh approach. This alone, without the discovery of America, or the invention of printing, or the collapse of the feudal system, would have eventually turned the course of civilization. Now the influence of relativity and quantum mechanics is beginning to be felt. Remember that these ideas are only about forty years old. The men who conceived them are still alive. It is difficult for us here to grasp their implications, but they are being taught in all our colleges, and a generation is growing up to whom these ideas are natural.

Another example is given by biology, and here again new concepts appeared at the beginning of this century. The discon-

tinuities in heredity called "mutations" appear due to changes in the genes which comprise the chromosomes in living cells. The genes are regarded as molecules, and the quantum mechanics of these astonishing molecules gives a glimpse of the secret of life itself. Yet this is no mechanistic view; it is rather a revelation which both humbles and exalts us with the profundity of creation. The philosophical implications of the new perceptions in biology inevitably involve the age-old problems of determinism and free will, the identity of God and man, and renew our belief in immortality.

The study of man has also progressed remarkably since 1900, in the fields which are identified by such overlapping terms as applied psychology, group dynamics, human relations and cultural anthropology. The difficulties in studying ourselves are formidable indeed. We must learn why individuals act as they do, what affects opinions and prejudices, what determines the personality of a nation, and how and why cultures interact in limiting or enlarging the possibilities of international cooperation. The anthropologists now tell us that there is nothing inherent in the nature of men which makes war either necessary or inevitable. It is a far cry, however, from the recognition of problems of human relations and patterns of culture to the development of significant improvements in them.

These few examples are sufficient to indicate, at least, that a revolution in thought is brewing, similar to but greater than the classical Renaissance. And as the stream of research adds to the reservoir of knowledge, it becomes more clear that this is a good time to be alive. Never before have such riches been promised in all fields of endeavor. Never before have men been more conscious of their abilities and shortcomings as architects of the future of mankind. No matter how vexatious today's problems may be, this is a time for optimism, for we can all participate in the opening act of the most glorious era in history.

DIESEL ENGINES OF THE GERMAN NAVY.

By E. C. Magdeburger, Member.

World War II was primarily a technological war, a war to determine which nation had the greatest ability to control the forces of nature and to organize means for wholesale destruction of its opponents. Human beings rose to sublime levels of new conceptions and sustained effort under the pressure of willpower generated by this emergency. It is therefore quite natural that hundreds of experts in scores of different lines of human endeavor were sent in the wake of the victorious allied armies to appraise and assess the effort made by the enemy. Their reports run into many thousands. Based on a study of such reports, on personal observation and on interrogation of many competent German engineers an attempt will be made to present a brief summary of the development of diesel engines for the German Navy during the last war.

Germany, the birthplace of the diesel engine, has maintained its enthusiasm for "the most efficient way of producing power by combustion" through the years since its inception and contributed generously to its development. The principal challenge and incentive to this development was the German Navy.

The Heritage of World War I.

There can be no doubt that the submarine supplied the most important impetus to development of diesel engines and vice versa, that the diesel engine was primarily responsible for the successful development of the submarine. The initial installations of diesel engines in German Submarines were made before the outbreak of World War I. Their operation was so successful that Admiral Tirpitz was willing to stake the final outcome of the war on "unrestricted submarine warfare". The German Navy was even so enthusiastic about diesel engines and their possibilities for propulsion of warships that orders were placed for two 10,000 horsepower engines of two-cycle double-acting type for battleship propulsion. Krupp and MAN each designed and built

one engine of this size and completed preliminary shop tests before the end of the war; MAN (Fig. 1) with greater success than Krupp. However, neither of these two engines survived the war and both were eventually scrapped.

The two MAN four-cycle submarine engines ((1) and (2) on Fig. 2) constituted the principal heritage of the first war. The U. S. Navy built a few copies of these engines under the name of "Bureau Type" at the Brooklyn Navy Yard (Fig. 3). These engines were of the air injection type and had oil cooled pistons. Cast steel cylinder housings and bedplates were the main features of their design, as compared with engines used in contemporary U. S. submarines, and accounted for the reduction in their weight and space requirements. These complicated steel castings were extremely difficult to produce and the Navy was never fully satisfied with the economics of manufacturing these diesel engines in a Naval Shipyard.

Diesel Engines for Pocket Battleships.

During the reconstruction period following the first World War the MAN Company abandoned air injection of fuel and developed a two-cycle engine with a very simple port-controlled "loop scavenging" system, although their wartime battleship engine used multiple-valve-in-the-head scavenging with all of its attendant complications. The simplicity of the new scavenging brought with it a sacrifice of considerable capacity to develop power since the superimposed rows of ports, exhaust above scavenging, covered over 30% of the piston stroke. With this arrangement of ports the volume of a fresh charge of air can not exceed 70% of the piston displacement and therefore the BMEP will not exceed 50 psi. A rotary valve in the exhaust passage was then added which improved the horsepower output by preventing the escape of any substantial part of the fresh air charge out of the cylinder through the exhaust port. This valve did not need to be very tight and therefore presented no maintenance problems.

Double-acting engines of this type (Fig. 4) were then built for propulsion of the three pocket battleships *Luetzow* (ex-*Deutschland*, ex-*Erstaz Preussen*), *Admiral Scheer*, and *Admiral Graf Spee;* two cruisers; *Leipzig* and *Nurnberg;* a gunnery training

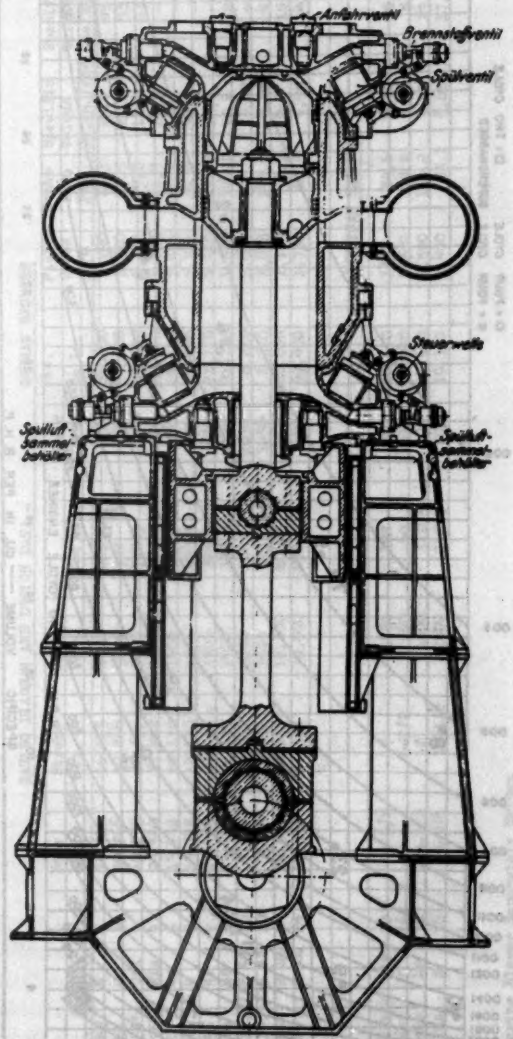


FIG. 1.—CROSS-SECTION OF MAN BATTLESHIP ENGINE OF WORLD WAR I.

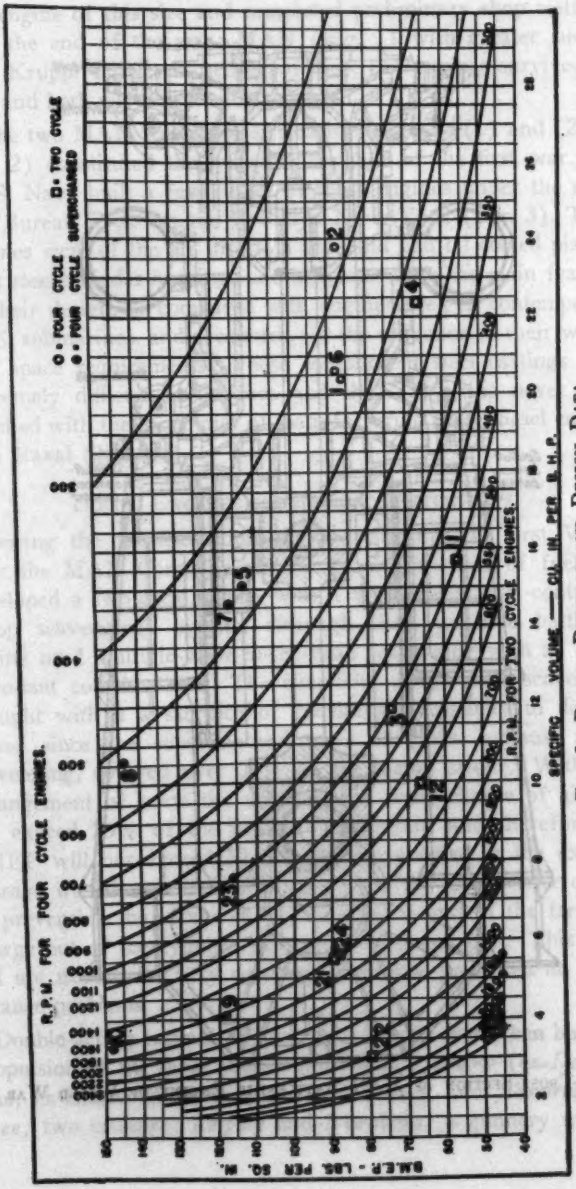

R.P.M. FOR FOUR CYCLE ENGINES.
O - FOUR CYCLE
● - FOUR CYCLE SUPERCHARGED
□ - TWO CYCLE
R.P.M. FOR TWO CYCLE ENGINES.
SPECIFIC VOLUME — CU. IN. PER B.H.P.
B.M.E.P. - LBS. PER SQ. IN.

FIG. 2.—RATING DIAGRAM AND DESIGN DATA.

RATING DIAGRAM AND DESIGN DATA –

GERMAN ENGINES

	Type	Cycle	Cyl.	Bore In.	Stroke In.	Brake H.P. Total	Brake H.P. One Cyl.	Speed RPM	Dis. Cu.In.	Piston Speed FPM	Brake MEP PSI	Specific Volume Cu.In./BHP	Specific Weight Lbs/BHP
1.	MAN 45/42	4	6	17.716	16.535	1200	200	450	24,500	1,240	86.3	20.4	49.6
2.	MAN 53/53	4	10	20.866	20.866	3000	300	380	71,400	1,320	87.5	23.8	53.33
3.	MAN 42/58*)	2DA	9	16.535	22.835	7100	788.9	450	84,600	1,713	73.8	11.9	17.64
4.	MAN 65/95*)	2DA	9	25.590	37.402	14,670	1630	265	329,500	1,655	66.5	22.4	---
5.	KRUPP 40/46	4S	6	15.748	18.110	1400	233.3	470	21,180	1,420	111.3	15.1	---
6	MAN 40/46	4	9	15.748	18.110	1535	170.5	435	31,770	1,313	88.0	20.6	---
7.	MAN 40/46	4S	9	15.748	18.110	2200	244.4	470	31,770	1,420	116.7	14.6	---
8.	MAN 40/46	4S	6	15.748	18.110	2000	333.3	520	21,180	1,570	144.0	10.6	19.25
9.	MB 501 18.5/25	4S	20	7.284	9.842	2000	100	1630	8,210	2,675	118.3	4.10	---
10.	MB 511 18.5/25	4S	20	7.284	9.842	2500	125	1630	8,210	2,675	148.0	3.28	4.0
11.	MAN 42/58	2 DA	24	16.535	22.835	14,400	600	450	226,000	1,713	56.1	15.7	15.38
12.	MAN 32/44	2 DA	24	12.598	17.323	10,000	416.7	600	99,400	1,732	66.5	9.94	14.3

AMERICAN ENGINES

21.	FM 38D-8 1/8	2CP	10	8-1/8	10	1680	168	800	10,370	1333	89.6	5.52	18.3
22.	GM 16-184A	2	16	6	6-1/2	960	60	1620	2,940	1755	79.8	3.06	5.7
23.	GM 16-278S	4S	16	9-1/2	12	1800	112.5	900	13,610	1,800	116.2	7.57	23.4
24.	GM 16-278A	2	16	8-3/4	10-1/2	1800	112.5	800	10,100	1,400	88.1	5.61	17.8

*) Scavenging Blower Separately Driven

ship, *Bremse;* and the airship *Graf Zeppelin.* The latter engine was never actually installed in the airship but was used in light marine installations. A four-cycle V-type Daimler Benz diesel engine, discussed below, was used instead.

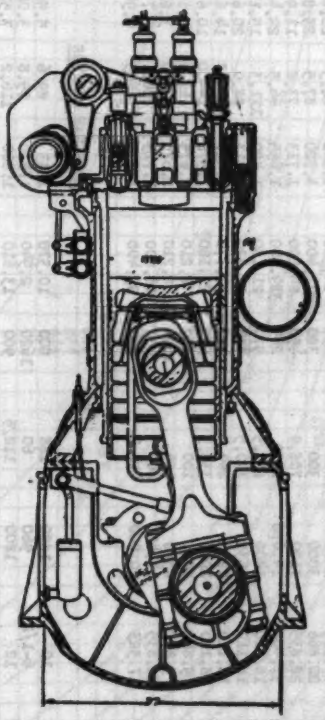


FIG. 3.—CROSS-SECTION OF "BUREAU TYPE" FOUR-CYCLE ENGINE.

Here again the U. S. Navy deemed it wise to foster the development of this type of engine in this country. The Hoover, Owens, Rentschler Co. (division of General Machinery Corp.), sole American licensee of MAN, developed one of the smaller sizes of this engine for propulsion of U. S. submarines and later for a number of 173 Ft. submarine chasers (PC Class). Without going into details it is only fair to record here that all of these U. S. submarines were re-engined during the war.

In 1936 the Bureau of Ships contracted with three engine

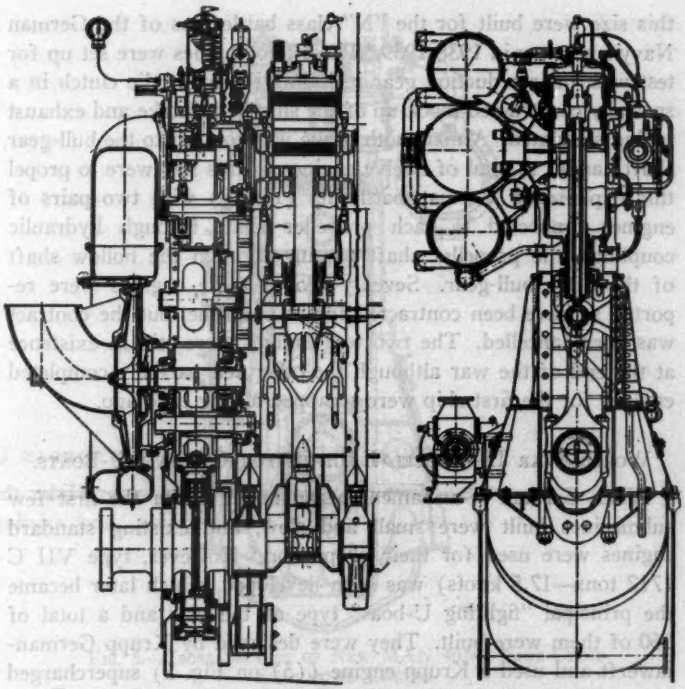


FIG. 4.—SECTIONS OF MAN TWO-CYCLE DOUBLE-ACTING ENGINE FOR "POCKET BATTLESHIPS."

builders, holding licenses from MAN, B&W and Sulzer, to build three-cylinder experimental engines of the same cylinder dimensions as the main propulsion engine of the pocket battleships ((3) on Fig. 2). Later two seven-cylinder Alco Sulzer type engines of these cylinder dimensions were built and installed in the tanker *Maumee*.

LARGEST ENGINES FOR BATTLESHIPS.

To offset the German pocket battleships the French Navy commissioned the MAN company to construct a three-cylinder experimental engine capable of developing 2000 horsepower per cylinder ((4) on Fig. 2). This engine was tested at the MAN plant in Augsburg and was delivered for further tests to the French Navy. Eventually, however, two nine-cylinder engines of

this size were built for the "N" class battleships of the German Navy sometime in 1938-1939. These two engines were set up for test with their reduction gear including the hydraulic clutch in a specially constructed mock-up of the ship with intake and exhaust piping simulated. A mammoth brake was coupled to the bull-gear shaft flange. A total of twelve engines of this size were to propel this triple-screw German battleship (Fig. 8) with two pairs of engines connected to each propeller shaft through hydraulic couplings, the propeller shaft passing through the hollow shaft of the after bull-gear. Seventy-two of these engines were reported to have been contracted for at one time, but the contract was later cancelled. The two test engines were still in existence at the end of the war although the other ten partially completed engines for the first ship were scrapped a long time ago.

World War II—Diesel Engines for German U-Boats.

When German re-armament began under Hitler the first few submarines built were small and slow, and existing standard engines were used for their propulsion. However, type VII C (712 tons—17.5 knots) was soon developed, which later became the principal "fighting U-boat" type of the war and a total of 660 of them were built. They were designed by Krupp Germaniawerft and used a Krupp engine ((5) on Fig. 2) supercharged by an engine-driven Roots type blower. Sixteen other firms (mostly shipyards) were given licenses to build these engines.

Concurrently a somewhat larger and faster type of submarine —IX C (900 tons—21 knots), of which a total of 146 were built throughout the war, was developed by Deschimag in Bremen. It was propelled by 9 cylinder MAN 40/46 engines ((7) on Fig. 2). A 6 cylinder engine ((8) on Fig. 2) of the same cylinder size was later used in the final pre-fabricated type XXI submarine (1600 tons, 17 knots on surface, 16 knots submerged), 119 of which were built but only one actually used in operations against the allies. The above three principal types of German submarines represent 80% of all the German submarines built (excluding midgets).

The development of the MAN 40/46 engine is international in character, beginning with a set of 8 cylinder engines for a Spanish submarine (Ech-21) in 1928. Two U. S. submarines

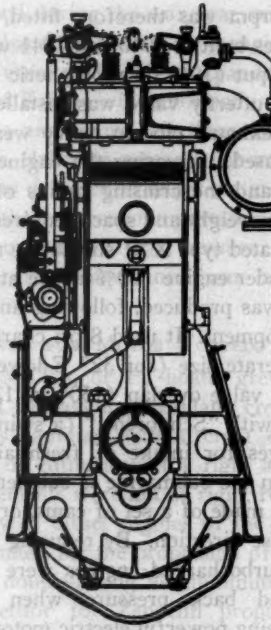


FIG. 5.—CROSS-SECTION OF NEW MAN 40/46 FOUR-CYCLE SUBMARINE ENGINE.

(*Cachalot* and *Cuttlefish*) were equipped with 9 cylinder engines of this type (Fig. 5 and (6) on Fig. 2). Two early German submarines (U-25 and 26) were propelled by 8 cylinder engines of this design and two others (U-27 and 36) had 6 cylinder engines. The first Büchi turbocharged (33% above atmospheric rating) 6 cylinder engines were installed in U-29 and uncooled aluminum pistons were first used in U-32. Seven cylinder engines of this type have been produced by Scotts in Greenock, England since 1938.

Turbocharging (40% above atmospheric rating) was added to the 9 cylinder engine ((7) on Fig. 2) used to propel the Type IX C submarines. However, war experience in submarines indicated that a turbocharged engine suffered an excessive loss of horsepower in heavy weather because of increased back pressure on the exhaust. A gear-driven centrifugal blower clutched in only

above 70% of rated rpm was therefore fitted, in place of the turbocharger, to engines built in 1940 and 1941 with a consequent loss of horsepower output (10% of atmospheric rating). In 1942 a manually operated butterfly valve was installed to control the back pressure in the exhaust pipe in heavy weather and turbochargers were again used, increasing the engine output and improving its efficiency and the cruising radius of the submarine.

An engine of smaller weight and space requirement was needed for the new pre-fabricated type XXI submarine and a new highly supercharged six cylinder engine (90% above atmospheric rating at 10% higher rpm) was produced, following an extended period of research and development. It used 8 psi charging air pressure with a cooler of moderate size (for 32° F lowering of air temperature) and greater valve overlap (150° vs 130°). To permit the engine to be used with "Schnorchel" (a swinging or telescoping mast with passages for intake of fresh air and discharge of exhaust gases when the submarine is submerged to periscope depth) clever use was made of a set of cams, originally provided for operation in reverse direction. By reducing valve overlap on the set of cams the turbocharged engines were made much less sensitive to increased back pressure when operating with Schnorchel.* There being powerful electric motors for submerged operation, all reversing was done on motors.

*Abstract of a routine shop test.					— Exhaust		Charg. Air	
	Test Hours	BHP-RPM	Turboch. RPM	Fuel Cons. Lbs/BHP/hr.	Temp. °F.	Back Pres. Ins. Water	Temp. °F.	Pres. Psi.
Rated Power	14	2000–522	12,240	.368	1080	16.5	167	8.0
Schnorchel ..	1	1400–470	10,000	.357	930	8.0	140	5.5

Lub. oil cons.—.008 lbs/BHP/hr. Lowest rpm—135 (50 BHP).

Structurally the difference between Krupp and MAN engines was in the engine housing which was welded in the latter while Krupp engines had cast steel housings. Inability to expand production of suitable steel castings, while plentiful facilities for steel fabrication had existed for many peacetime applications

which were curbed during the war, was said to have been one of the reasons for abandoning the Krupp engine.

Production of crankshafts when expanded beyond the facilities and experience of Krupp, which normally supplied them, resulted in slag inclusions and many rejections and necessitated a two-piece design. An elaborate research laboratory for testing full-size crankshafts to destruction under static load and from fatigue established later the relative unimportance of many types of slag inclusions. Throughout the war, however, the supply of diesel engines for German submarines was adequate to meet all requirements.

DIESEL ENGINES FOR MOTOR TORPEDO BOATS (E-BOATS).

The wartime MTB's, or PT-boats, grew out of crash-boats, which were used for rescuing seaplane crews and were manned by ground crews familiar with aircraft engines. Furthermore the brief period of enthusiasm for rigid airships, where weight and space requirements were not as rigid, resulted in the development in Germany of diesel engines of the "airship type", which were eminently suited for the propulsion of the MTB's. Both the MAN two-cycle double-acting all-aluminum engine and its more successful competitor for Zeppelin propulsion, the four-cycle V-type Mercedes-Benz diesel engine, were originally used in MTB's. However, experience soon demonstrated that the Mercedes-Benz engine was superior to the MAN type in this application and model MB 501 soon became the standard propelling plant for German E-boats.

MB 501 is a 20 cylinder engine (40° V) (Fig. 6), each bank having its own camshaft and exhaust header on the outside and a common intake manifold in the center ((9) on Fig. 2). This design is a modification of the original MB 502 installed in airships and the first E-boats, which is a V-16 (50° V) with cylinders of slightly smaller bore and stroke (175 x 230 vs 185 x 250) and a common camshaft for both banks in the center. The MB 501 was later improved by the addition of a gear-driven (9.3 to 1 speed ratio) centrifugal supercharger (25% above atmospheric rating) capable of compressing to 6½—8 psi gage, with an air cooler reducing air temperature by 60-70°F. The improved model was designated MB 511 ((10) on Fig. 2). These conversions were made on service engines being returned for overhaul.

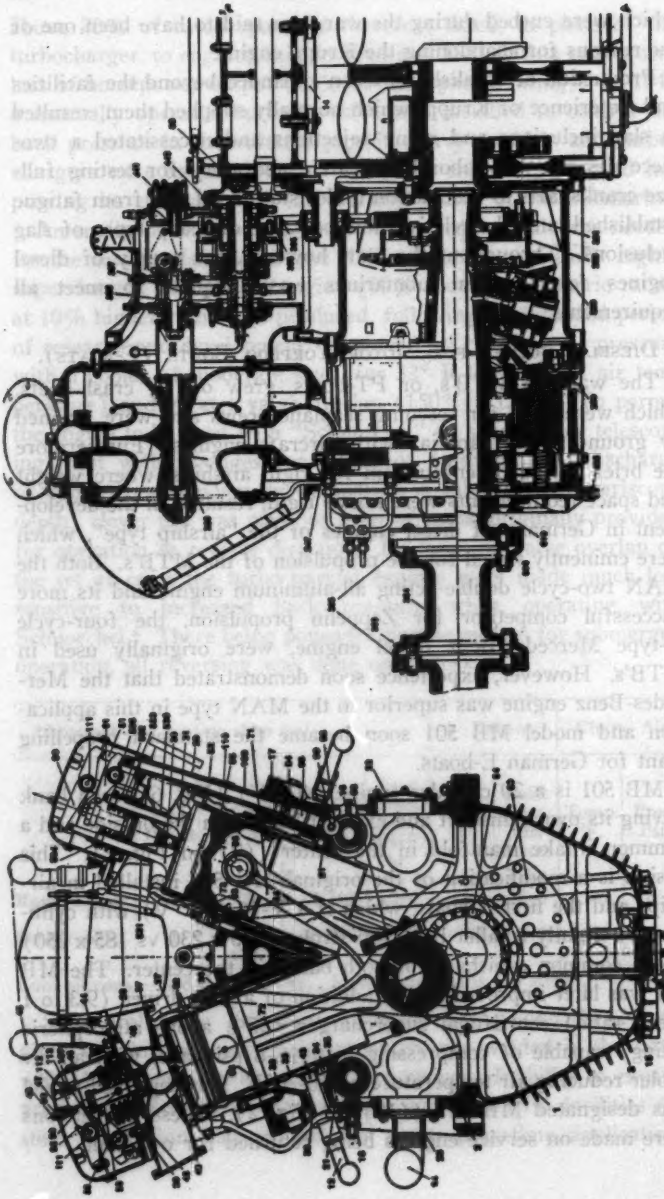


FIG. 6.—SECTIONS OF DAIMLER-BENZ FOUR-CYCLE SUPERCHARGED ENGINE.

The original MB502 had roller bearings on both main journals and crank pins with serrated-split outer races. Connecting rods of fork and blade type were used with three rows of rollers in the bearing. Eventually much trouble from cracking of rollers, even after only a short period of operation, forced a sleeve bearing to be developed to replace the roller bearings on crankpins, although on main journals they have never given any trouble and remain there now to contribute to the tight cylinder spacing, compactness and light weight of the engine.

The crankcase and bedplate are aluminum castings, split along the centerline of main bearings and held together with tie rods to take care of tension stresses. Individual hollow forged cylinders with closed ends are flanged in the middle where piston thrust is greatest. A light steel jacket, valve ports and top structure are welded to the cylinder. Forged aluminum pistons are used. The engines are directly reversible with a 1.72 - 1 reduction gear of bevel planetary type including a disc type clutch attached. The well-known Benz precombustion chamber and Bosch injection pumps are used.

It is of interest to note that an attempt to use a Büchi turbocharger in these engines was made some time ago and failed after it was established that back pressure waves interfered with proper scavenging and consequently the output of several cylinders. Two standard models of turbocharger were used, mounted on the reduction gear housing, and four separate exhaust manifolds (2 and 3 cylinders for each) for each bank. It was decided that the problem was unsolvable under circumstances existing in wartime since any other location of turbochargers would increase the outline dimensions of the unit, and that was unacceptable.

An attempt was made to meet the insistent demand of the German Navy for engines of greater output by increasing both BMEP and rpm. Charging air pressure was raised and precombustion chamber volume was increased to get higher BMEP, while moving parts were lightened to permit higher rpm. A new model—MB518—was to develop 3000 BHP at 1720 rpm with a BMEP of 166 psi and a few engines of this type were built for test. However, piston failures due to cracks in the head were not eliminated by any designs tested at the above rating up to the end

of the war even though the addition of cooling by oil spraying had a very beneficial effect.

Diesel Engines for Surface Warships.

As in World War I so in the last war the German enthusiasm for diesel engines, or perhaps more accurately, the "pinch of fuel shortage", finally resulted in the construction of a very large unit suitable for battleship propulsion.

The success of the Mercedes-Benz V-type engines and the building of hundreds of V-type two-cycle engines by General Motors in this country (of which the Germans, no doubt, were well informed) brought about the conversion of MAN and resulted in a Vee engine of two-cycle double-acting type with the basic cylinder design duplicating that used for the pocket battleship engines ((11) on Fig. 2). An order for 48 engines of this model for the propulsion of "O" type battleships (Fig. 9) is reported to have been placed with MAN and one engine is definitely known to exist while others were only partially assembled. This engine completed a shop test of 300 hours. The rest of the order was cancelled.

In the 1942 program one 2656 ton full load displacement destroyer (Z-51) (Figs. 10 and 12) was included, which was designed for propulsion by six 10,000 horsepower diesel engines, (Fig. 7). Four of these engines ((12) on Fig. 2) were to drive the center shaft through a common reduction gear and individual hydraulic clutches similar to pre-war drives of pocket battleships and cruisers. Each pair of engines and the reduction gear with clutches were in separate rooms. The remaining two engines were direct coupled to the wing shafts and their engine room was separated by a short auxiliary engine room from the after pair of engines of the center shaft. Other auxiliaries were arranged above the reduction gear. The Germans later designed a larger type of destroyer (Z52-59) (Fig. 11), in which eight of these engines were to be used.

The main diesel engines are of the MAN two-cycle double-acting V-type with a geared centrifugal blower above the crankshaft driving flange and attached pumps. Of outstanding interest is the installation of exhaust gas driven turbo-blower in the superstructure above the main deck (one for each engine) dis-

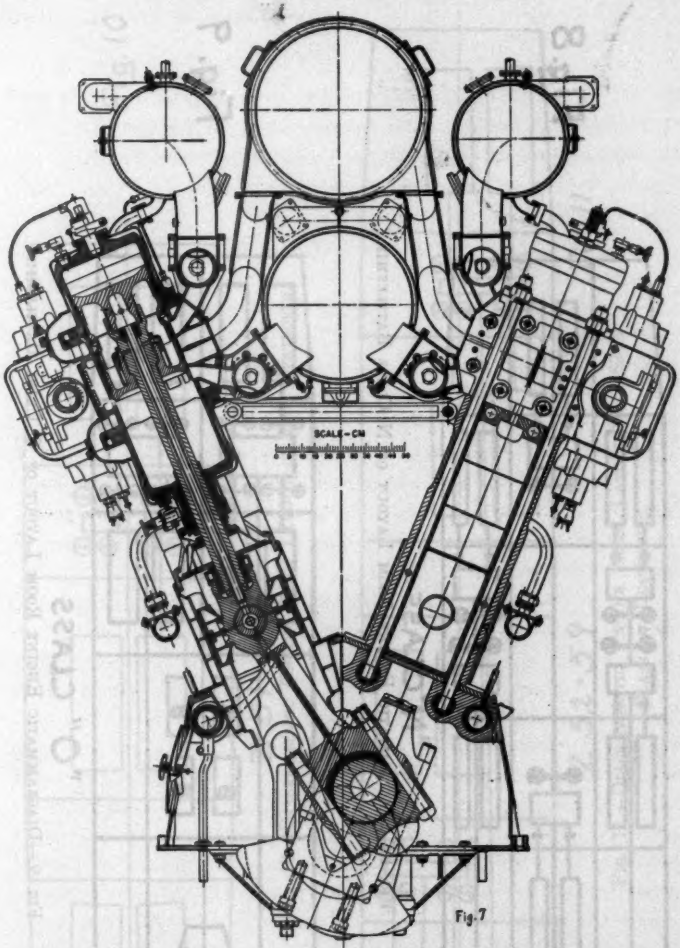


FIG. 7.—CROSS-SECTION OF MAN V-TYPE ENGINE FOR DESTROYER.

charging into the engine driven blower and thereby forming a two-stage system of compression without intercooling for the charging air. At low speeds most of the compression is done by the geared blower because the temperature of exhaust gases is low while at higher-speeds the geared blowers are relieved of all work

Fig. 8

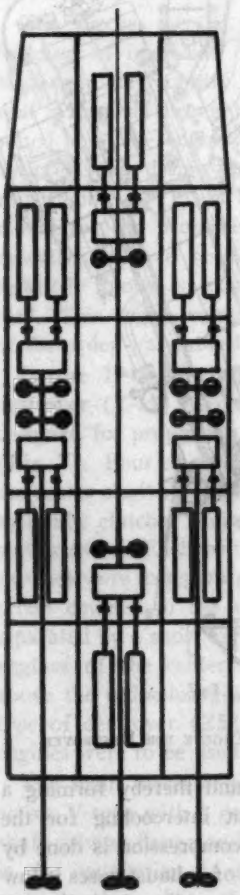


Fig. 8.—Diagrammatic Engine Room Layout of N Class Battleship.

Fig. 9

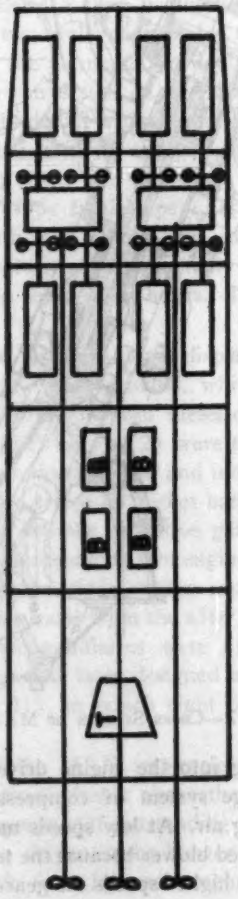


Fig. 9.—Diagrammatic Engine Room Layout of O Class Battleship.

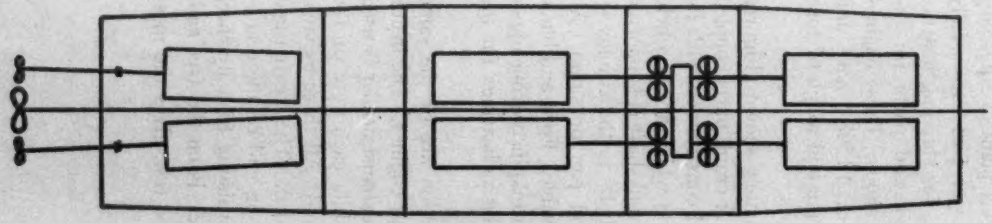


Fig. 10

FIG. 10.—DIAGRAMMATIC ENGINE ROOM LAYOUT OF Z-51 DESTROYER.

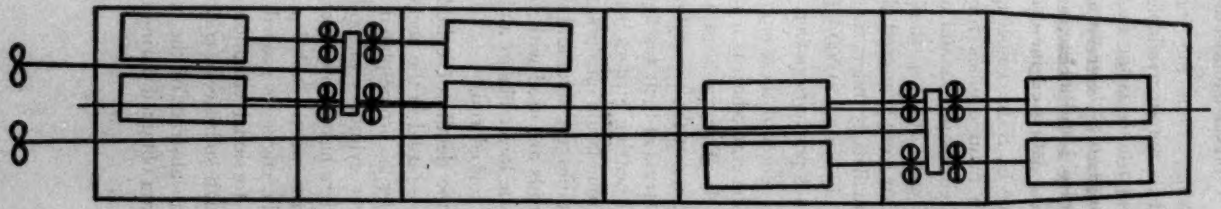


Fig. 11

FIG. 11.—DIAGRAMMATIC ENGINE ROOM LAYOUT OF Z-52-59 CLASS OF DESTROYERS.

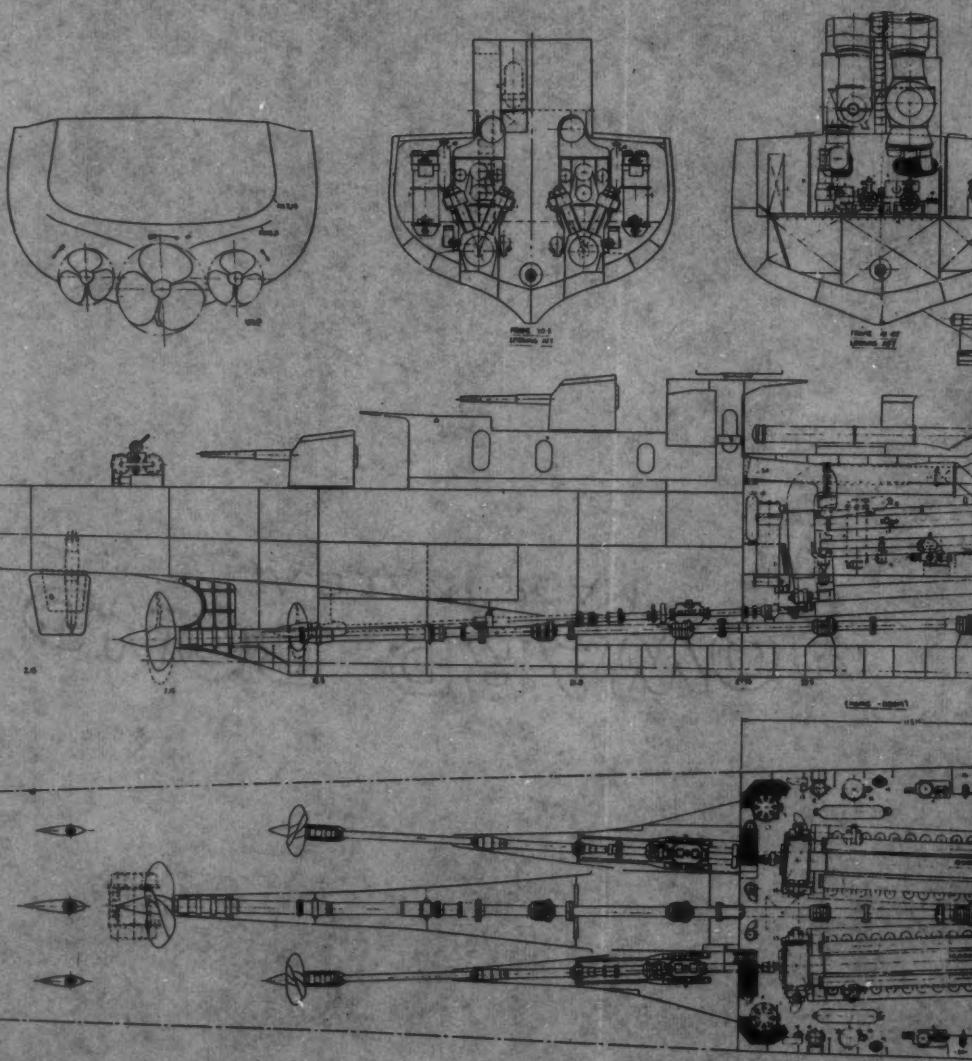
of compression. Thus the fuel-consumption of the engine is lowered and its output available for propulsion is increased. Of particular importance is the fact that such a system is applicable to all existing two-cycle engines by merely substituting a divided exhaust manifold and adding a turbocharger of appropriate size ahead of the existing engine driven scavenging blower.

At the time U. S. forces occupied the MAN plant in Augsburg three of the main engines for this destroyer had successfully completed their shop tests and were waiting for shipping instructions, the fourth stood partially assembled, and parts for the remaining two were scattered through the shops. These engines are of a smaller cylinder size (32/44) or 12.598″ bore and 17.3238″ stroke, rated 10,000 BHP at 600 rpm and weigh 67 tons or 14¾ pounds per horsepower.

The construction of these engines is adequately shown (Figures 3 and 4). The crankshaft is made of two six-crank parts coupled in a crankweb. The crankcase is of welded construction with tie rods holding down the two cylinder supports which are also held together by struts at their top flange. The crankshaft is supported by cast steel bearing girders held by long bolts to the crankcase. The ships foundation girders form the oil pan or sump. All crosshead guides are alike and easily removable. Fork and blade connecting rods are used with removable crankpin bearing upper half-shells, the lower blades and caps serving as bearings for the caps of the forked rod.

Most of the details of cylinder construction are of the same design as those used in the pocket battleships engines. The inside of the cylinder is chrome-plated. A central exhaust header is used for the lower cylinders of both banks, while each row of the upper cylinders has its own exhaust manifold. All three exhaust manifolds are water cooled and subdivided. The centrifugal blower of the engine is geared to the driving end of the crankshaft and a torsional vibration damper, consisting of a flywheel driven through split-sleeve spring assemblies, is on the free end.

Table I, an abstract of the report of type test, gives the most important data obtained during these tests.



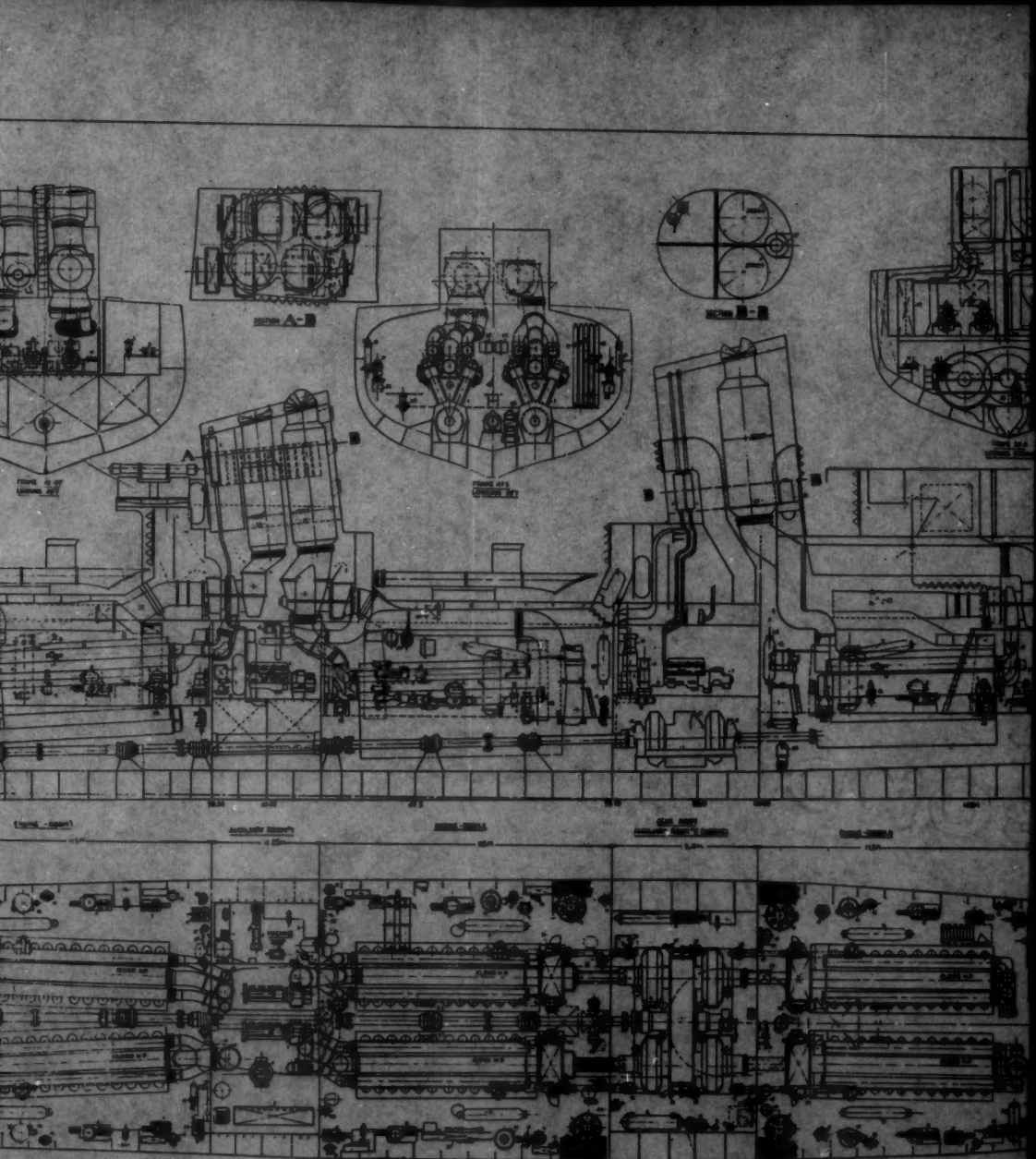


FIG. 12.—MACHINERY ARRANGEMENT PLAN Z-51 DESTROYER.

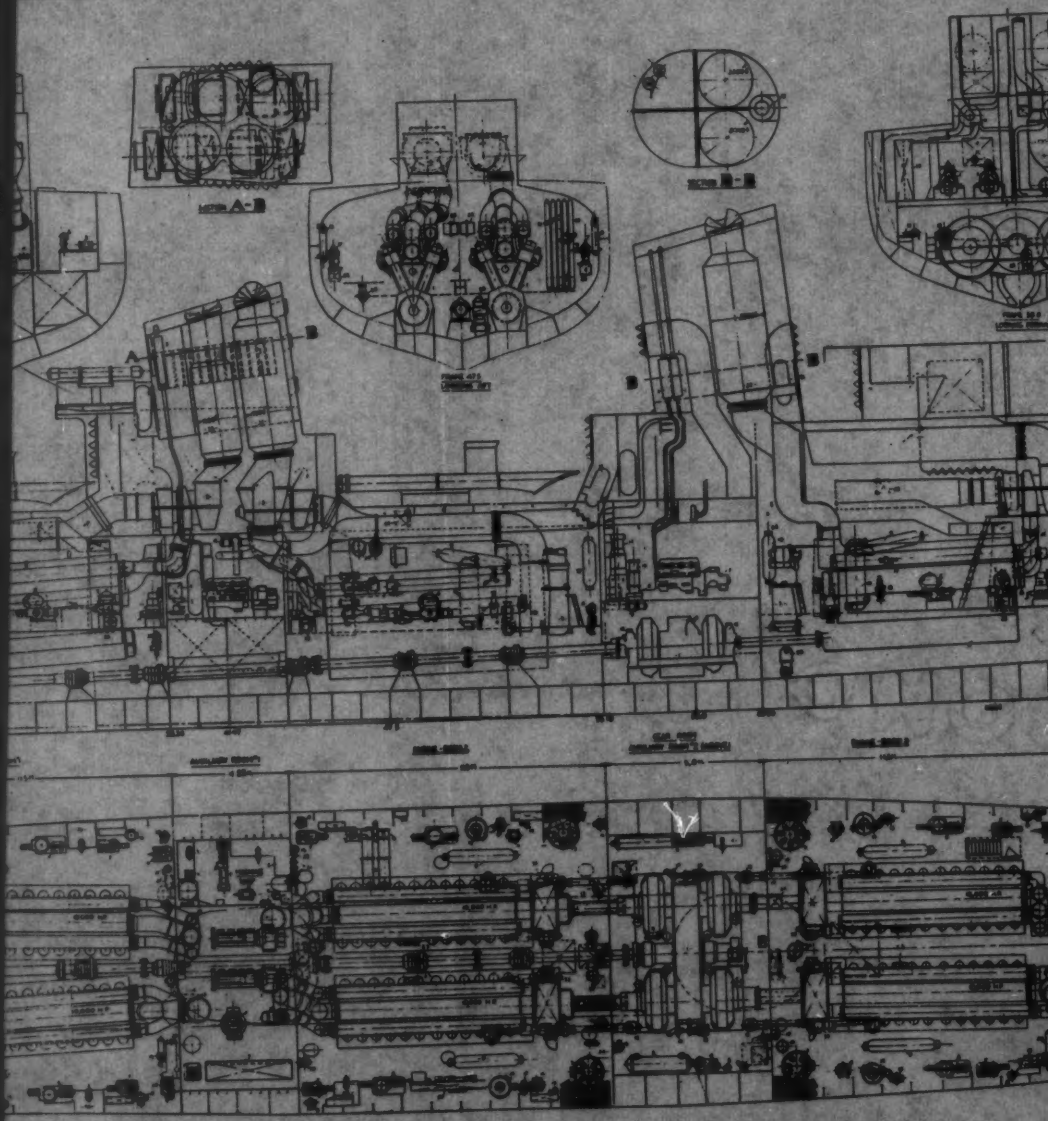


FIG. 12.—MACHINERY ARRANGEMENT PLAN Z-51 DESTROYER.

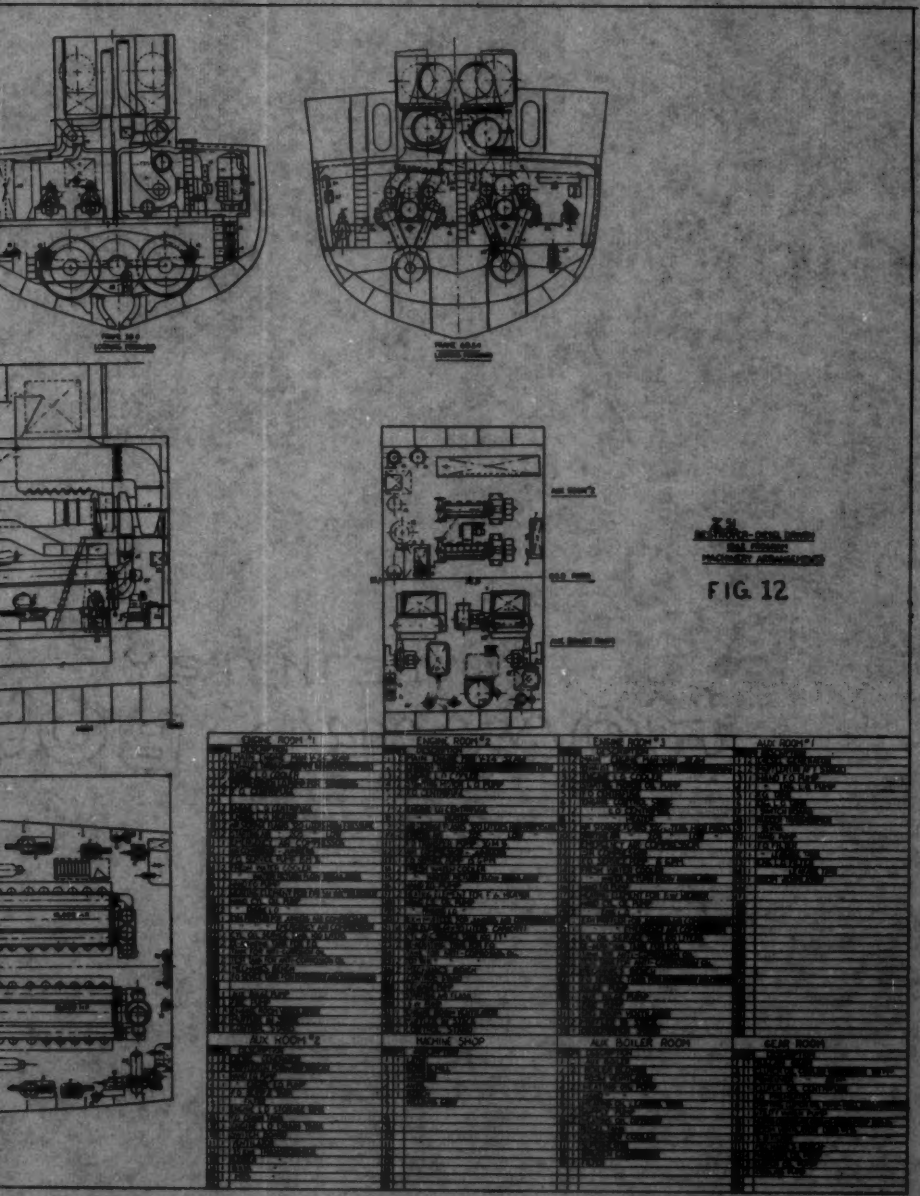


FIG. 12

Table I

ABSTRACT OF REPORT OF TYPE TEST OF MAN 24 CYLINDER
TWO-CYCLE DOUBLE-ACTING DIESEL ENGINE, MODEL V122 32/44, RATED 10,000 BHP AT 600 RPM

Dates of test 8-12 Feb. 1944
Average values of the most important data obtained

	AHEAD				ASTERN		OVERLOAD xx)		
Load	1/4 power	1/2 or cruising	3/4 power	Rated power	Rated power	Maximum power	110%	120%	125%
(No. of engines operating)	(one engine)	(2 engines)	6	6	6	6	6	6	6
Length of test - hours	1-1/4	2-1/2	1-1/2	72	1	1/2	2	1	1/4
RPM	225	326	542	600	399	500	600	600	600
BHP	923x)	3,460	7,510	10,000	3,990	5,000	11,000	12,000	12,500
BMEP - psi	16.1	41.8	54.4	65.6	39.2	39.2	72.0	78.8	81.7
Torque in percent rated	24.5	63.5	83.0	100	60.0	60.0	110	120	125
Throttle position	3.2	5.4	7.05	8.4	5.3	5.6	8.0	8.9	9.0
Fuel consumption lbs/BHP/hr	.556	.417	.416	.418	.430	.453	.418	.419	.422
Exhaust temperature - upperleft	241	406	527	633	531	592	680	734	752
in manifold - °F - upperright	241	403	523	626	516	576	680	716	734
- lower	165	324	484	610	428	509	684	734	752
Peak pressure upper cyl.	582	768	945	966	718	682	973	973	960
p.s.i. lower "	469	640	875	895	610	610	910	896	888
Exhaust at all loads only slightly visible. ---------- slightly visible								moderately visible	-------
Lubricating oil consumption at rated power ------.00242 lbs/BHP/hr ----------									

x) Brake would not absorb greater load at this speed.
xx) Fuel pump plungers of larger diameter were then installed and additional tests of 10,20 and 25% overload were run on August 2, 1944.

THE POSITIONING OF PROPELLERS AND SHAFTS.

By N. J. Brazell (Member).

The formula for battle victory prescribed by General N. B. Forrest, of the Confederacy, in his epigram "Get there fustest with the motest" was valid throughout the military and naval actions of World War II; from Warsaw to Berlin and from Pearl Harbor to Tokyo.

In order to comply with the postulate of General Forrest, at any geographical location on the globe, it is essential to provide at least a twenty per cent increase in the full power speed of present Naval auxiliary ships as well as vessels comprising the protective screen of a Task Force. It is apparent that higher sustained sea speed may be obtained through reduction in the relative hull resistance per unit of installed power, or by increasing the thrust which is developed for propulsion. A preliminary calculation indicates that recourse to a combination of both methods will be necessary in order to economically obtain the additional speed required. A first estimate indicates that the installed S.H.P. of the power plant must be doubled. This estimate gives consideration to the savings in displacement which may presently be achieved through the sound employment of light weight materials and high strength alloys.

It is emphasized that doubling the installed S.H.P. does not necessarily assure the actual doubling of the propulsive thrust developed by the propeller, especially since it appears desirable to make no change in the number of propeller shafts installed in the new ship. Since the objective of a twenty per cent increase in ship speed requires a doubling of the thrust output of the propeller, it is manifest that the task becomes basically a propeller design problem, and this problem is not a minor one. For instance, the installed S.H.P. of the Mahan destroyer class was double the power harnessed in the old "Four Piper" destroyers of World War I design. The propeller design difficulty encountered on the Mahan was finally resolved, but only after a comprehensive series

of propeller model tests in the new twelve inch variable pressure hydraulic tunnel. Now it is proposed to once again double the demand upon the propeller. Despite the fact that we now have the new NACABS propeller design, which embodies cambered hydrofoils, and resists the incidence of laminar cavitation, the cooperation of all is necessary in order to afford the propeller an opportunity to produce the thrust which is required for obtaining increased speed.

During the preliminary design stage of new ships, it is essential for the Naval Architect to fully understand the environment which aids propeller efficiency, and particularly the consequence of any and all hydrodynamic restraints, which are often embodied in a new design and adversely affect propeller performance. A review of the principal factors, which determine the interaction between the hull of a ship and the screw propeller, therefore, appears to be timely. This paper will survey the variation in overall propeller performance occasioned by the positioning of propellers and the alignment of shafts, with the objective of appraising the modern trend in the achievement of superior ship propulsion.

General Considerations.

The detrimental consequence of the thrust deduction coefficient may have provoked the remark that "It would be desirable to drive a Liverpool bound liner by propellers positioned in the Pacific Ocean". It is, of course, necessary to surround the propeller with all of the operating safeguards which sound engineering can employ. The hull body of a ship affords a natural means for protecting the propeller, and hence, the propeller should be located at the stern of the ship and as close to the hull as acceptable vibrations will permit. Since bow propellers cannot be adequately protected, other unfavorable aspects of traction drive for propelling ships will not be discussed.

There is no justification for the prevailing tendency to position propellers in a so-called "favorable wake". The usual wake distribution over a propeller disc imposes a load limitation which becomes a liability rather than an asset. It should be borne in mind that the thrust deduction coefficient is rigorously interrelated with the potential or stream-line wake fraction. A similitude with the axiom of Leonardo Da Vinci, explaining the action of the

simple lever, would be: "What is gained in potential wake, is lost in thrust deduction." And actually the profit from the frictional and wave wake components is generally exceeded by the loss which is occasioned by the wake fraction demanding a smaller propeller pitch ratio, resulting in a less efficient propeller which is forced to operate at increased slip.

A fore and aft disposed contra-vane or even a simple skeg will reduce the natural loss in efficiency occasioned by the rotation of the propeller slip stream and therefore skeg supported propeller shafts are often advantageous. Propulsive coefficients, in the "high 80's", which may be obtained in single screw ships, are a direct result of the elimination of slip stream rotation which is achieved by the directional flow effect of the centerline skeg and rudder. The opinion that a high P.C. is produced by a high wake fraction is false.

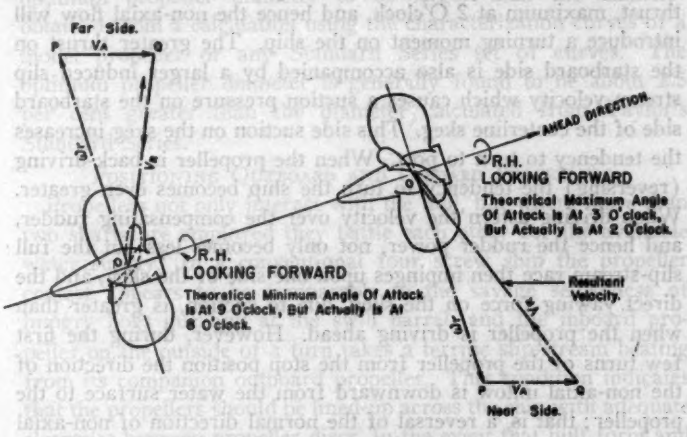
In order to obtain security for the ship, it is essential to protect the propellers from major casualties and damage. It is, therefore, evident that the propellers should be positioned not only in way of the stern of the ship, but also as close to the hull as possible. But in order to obtain smooth operation, the clearance between the hull and the tips of the screw propeller blades must be of sufficient magnitude to avoid the excitation of objectionable vibrations. A method for determining the amount of working tip clearance for a propeller will be discussed later.

Non-Axial Flow.

The screw propeller is a thrust instrument which obtains the thrust force required for propulsion by the principle of reaction (Newton's 3rd Law of Motion). And further, the screw propeller with its shaft is an axial flow mechanism and its capacity to deliver a thrust force is higher when the stream lines in the induced slip stream are axially disposed. The consequence of non-axial flow in highly loaded propellers is not generally appreciated.

The flow of water to the propeller is never in a horizontal axial stream line. At the stern of a ship the direction of the water particles is upwards and inwards—which the propeller designer terms "non-axial-flow". For instance, when water is flowing upwards from keel to propeller disc the consequence is to increase

the hydrodynamic angle of attack at one position during rotation, which in turn momentarily increases the thrust loading on the propeller blade. The effect of non-axial flow occasioned by a raked shaft is shown in Figure 1. The velocity vector, *wr*, drawn



PROPELLER "NON-AXIALFLOW"
VELOCITY LINE VECTOR DIAGRAM

FIG. 1

perpendicular to the centerline of the shaft is the tangential velocity component due to the rotation of the propeller. Then V_A represents the speed of advance of the propeller parallel to the surface of the water. The velocity vector, V_R, is the resultant of these two velocities. Since the propeller is a reaction thrust instrument the direction of the resultant, V_R, is reversed in direction as shown, but the magnitude remains the same. This line vector diagram, Figure 1, complies with the conventional method of propeller graphics. It will be noted in Figure 1 that the magnitude of the resultant, V_R, and the angle of attack are greater for the blade on the "near" side (3 O'clock) and substantially less on the "far" side (9 O'clock). Observation in the test laboratory shows that the maximum thrust load imposed upon a right hand turning propeller blade in upward non-axial flow actually takes place at the 2 Oclock position. The transient and intensive peak propeller blade loading at the 2 O'clock position results in an earlier inception of cavitation.

Non-axial flow causes a single screw ship fitted with a right hand rotating propeller to turn to port when driving the vessel ahead. Since the general direction of flow is upward to the propeller disc there will be produced a non-symmetrical propeller thrust, maximum at 2 O'clock, and hence the non-axial flow will introduce a turning moment on the ship. The greater thrust on the starboard side is also accompanied by a larger induced slip stream velocity which causes a suction pressure on the starboard side of the centerline skeg. This side suction on the skeg increases the tendency to turn to port. When the propeller is back-driving (reversing) the tendency to turn the ship becomes even greater. When driving astern the velocity over the compensating rudder, and hence the rudder power, not only becomes less but the full slip-stream race then impinges upon one side of the skeg, and the direct yawing force on the skeg, due to suction, is greater than when the propeller is driving ahead. However, during the first few turns of the propeller from the stop position the direction of the non-axial inflow is downward from the water surface to the propeller; that is, a reversal of the normal direction of non-axial flow results in an opposite yawing moment which will momentarily exist until the natural induced slip stream is established. Hence the ability to "kick over" the stern of a single screw ship during a backing and filling maneuver.

The effect of non-axial flow was evidenced in a recent standardization trial of a destroyer. At full power and equal R.P.M. the starboard shaft absorbed 4.7 per cent more power than the port shaft. The shaft spread angles (outboard) were equal (9½ minutes), but the rake angle of the port shaft was 1°-30′ larger than the rake angle of the starboard shaft (4°-4′ vs. 2°-34′). The two propellers were manufactured at the Philadelphia Naval Shipyard on the Morton Profiling Machine using the same model, and consequently the right hand and the left hand propellers were dimensionally similar. It is apparent that non-axial flow will adversely affect propeller performance in the field of cavitation. In order to avoid non-axial flow it is evident that the shaft rake and spread angles should be minimized. This requirement, in addition to propeller protection, calls for positioning the propeller as close to the hull as practicable. It should also be noted that

excessive propeller tip clearances demand larger shaft struts and also entail increased appendage resistances.

The universal existence of non-axial flow in way of the propeller aboard ship is the primary reason that tests have shown the optimum propeller diameter to be larger than the diameter obtained from a calculation using the characterization curves of a model propeller or any Standard Series set of curves. The optimum propeller diameter is generally found to be about 2.5 per cent greater than the diameter calculated from Taylor's Standard Series.

Positioning Outboard and Inboard Propellers.

Propellers not only interact with the hull, but where more than two shafts are employed they battle each other for the available water supply. In a conventional four screw ship the propeller contest appears to be comparable to the savage selfishness of hungry hogs guzzling at the swill barrel, and the inboard propeller on the outside of a turn takes a terrific slip-stream beating from its companion outboard propeller. This condition indicates that the propellers should be lined-up across the ship with adequate clearance between propeller discs. In the event that hull form and battle damage integrity of the machinery spaces preclude athwartship alignment of propellers it is desirable to separate the forward from the after propeller by a minimum of five propeller diameters. The vertical height of the forward propellers above the after propellers should be at least ten per cent of the propeller diameter, but with attention given to affording sufficient propeller tip immersion when the ship is rolling in a seaway.

The huge shaft horsepower output per shaft presently being installed has resulted in large propeller diameters and has caused the propeller tips to project almost five feet below the base line in certain destroyer applications. The available drydocking facilities will probably permit an additional two feet and the destroyer of the near future may have propellers projecting seven feet below the protecting keel line of the hull. Drydocking such a destroyer will be something to see, and the directional stabilizing effect of large propellers positioned in this manner is certain to impair ship maneuverability and to increase the tactical diameter.

Shaft Struts and Bosses.

The simplest method of supporting an outboard propeller shaft

is by means of conventional struts. Care must be exercised in positioning the forward or spring strut in order to avoid probable strut arm failure during a hard turn. The lateral hydrodynamic resistance to turning is much less with the cut-away and flat transom stern of a destroyer than in a normal hull form. The transverse thrust from a hard-over rudder, at full speed, tends to deflect the ship's after body. But the static and dynamic stiffness of the rotating shaft, which is augmented by the inertia of the propeller slip stream column, affords the principal structural restraint of the rudder force. And so it may be said that a destroyer physically maneuvers about her propeller shafts with the after stern tube bearing acting as the fulcrum of the restraining lever. Incidentally, this is the reason why the after stern tube bearing has always been a maintenance problem in the destroyer and light cruiser class vessels.

The three bearing and the four bearing arrangement of water lubricated bearings is shown in Figure 2. The three bearing arrangement is preferred where practical. In order to raise the critical flexural whirling speed of the shaft well above the running range, the span ratio L_2 over L_3 (See Figure 2) is sometimes made as low as .85. This arrangement may invite trouble whereas

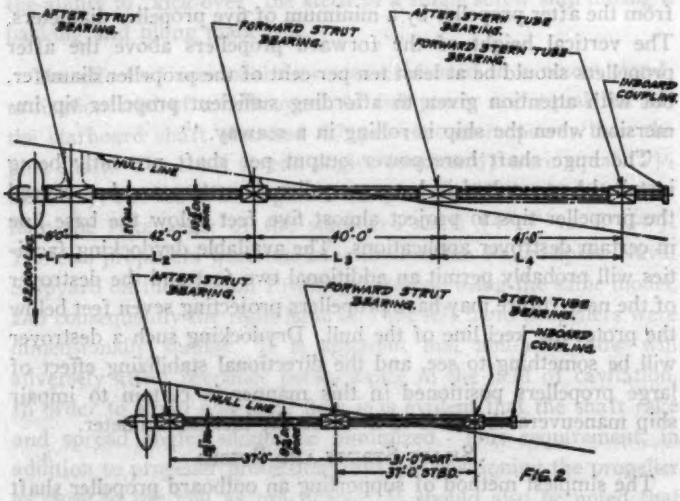


FIG. 2

an L_2 over L_3 span ratio slightly greater than unity is generally safer and better practice. It is of the utmost importance to insure an adequate positive bearing pressure upon the lower bearing staves of the forward strut; especially in high-powered ships.

In a recent destroyer design, having outboard shafting 16.5" in diameter, the computed total net static bearing load upon the spring strut was about one ton and, when the ship was underway, the direct dynamic lifting force on the exposed raked propeller shaft materially reduced the one ton positive static bearing load. The actual load upon the spring strut bearing was further reduced by the *magnus-effect* (resulting from the rotating shaft being positioned in the outward-flow pattern caused by the induced tangential (rotational) velocity in the slip stream occasioned by outboard turning propellers). This initially light static bearing load, which was reduced by the hydrodynamic lift, caused the shaft to float within the forward strut bearing so that the virtual bearing span became $L_2 + L_3$ (see Figure 2). Hence the critical flexural whirling speed of the shaft was brought within the operating range of RPM and when the vessel was put into hard turns. The anti-node of the whirling shaft, in way of the spring strut, hammered the bearing with a force of such magnitude that the single arm supporting the forward strut was broken. It was necessary to replace the single arm of the forward strut with a strong double arm support to the hull. The single arm spring strut operated successfully on an earlier class of vessels, where an equal SHP was transmitted at slightly higher RPM with identical shafts and bearing spacing, but in this latter instance the weight of the overhanging propeller was about 2000 pounds less. The 2000 pounds extra weight of the overhanging propeller caused a change in the static deflection curve of the shaft and reduced the bearing load on the forward strut to such an extent that a major casualty resulted.

The method of moment distribution is a convenient means for calculating static loads upon water lubricated bearings where the propulsion shafting forms a continuous beam (i.e. more than two shaft bearing supports, which is the usual case). The customary practice of employing conventional formulae which are based upon the Theorem of Three Moments applied to the three after shaft spans has resulted in much confusion. This perplexity was

occasioned by errors made in the algebraic change of sign subsequent to the addition, subtraction, multiplication, and division operations which are required in the solution of the three moment formulae. It is often desirable to accurately determine the loads upon all of the bearings including the line shaft spring bearings. Since the propulsion shaft may form a continuous beam consisting of seventeen bearing supports it is evident that recourse to the Theorem of Three Moments would entail a tremendous amount of computation. By contrast, all of the seventeen bearing loads can be calculated within one working day by means of the method of moment distribution.

In order to preclude future casualties resulting from a critical flexural shaft whirling speed with the anti-node in way of the forward strut two paradoxical design provisions are essential. First, the bearing span adjacent to the overhanging propeller should be of generous dimensions—at least twenty shaft diameters. And second, the length of the conventional spring strut bearing should be shortened, that is, not to exceed one and one half shaft diameters. And further, a static bearing load of at least twenty-five pounds per square inch of bearing area should be provided for the spring strut bearing. As a matter of fact, the excessive length of all present water lubricated bearings imposes a restraint upon the natural tendency for minor shaft deflection during maneuvering which in turn results in localized wear and premature failure of the bearing wearing material. There exists a pressing need for the development of a water lubricated bearing having continuous film lubrication and a non-swelling bearing material which will permit the length of the stern tube and after strut bearings to be equal to or less than three times the diameter of the shaft sleeve. Strut and stern tube bearings in excess of fourteen feet in length, which have been installed for propeller shafts two feet in diameter, are the result of extrapolation rather than the consequence of sound engineering.

The web member of spectacle frames and bosses which are sometimes used to support the outboard propulsion shafts affords a contravane effect in the upper arcs of the propeller disc and therefore appears to be advantageous. But the disposition of the web inclination to the hull can be set to parallel the lines of flow for one and only one speed. If a cruising speed is also an essential

the web structure will necessarily cause interference and increased resistance at either the full power or the cruising condition. Hence webs and bosses are not desirable for naval applications. It is also apparent that a spectacle frame structure is initially more costly than the conventional shaft strut installation.

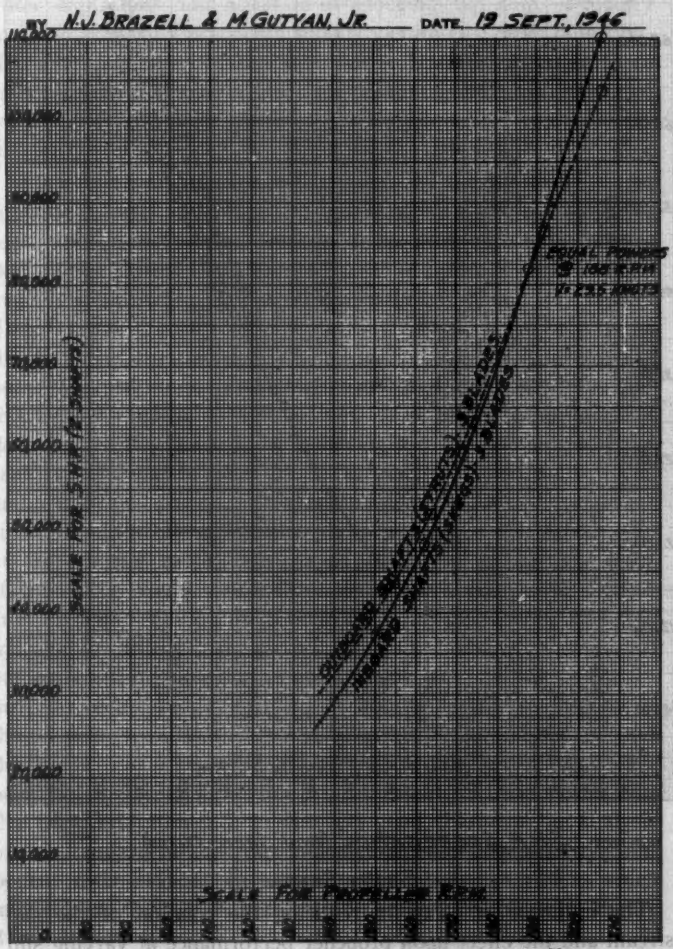

BY N.J. BRAZELL & M. GUTYAN, JR. DATE 19 SEPT., 1946
SCALE FOR SHAFT (2 SHAFTS)
SCALE FOR PROPELLER RPM
SUBJECT: S.H.P. SEGREGATION (TWIN SKEGS INBOARD)

Fig. 3

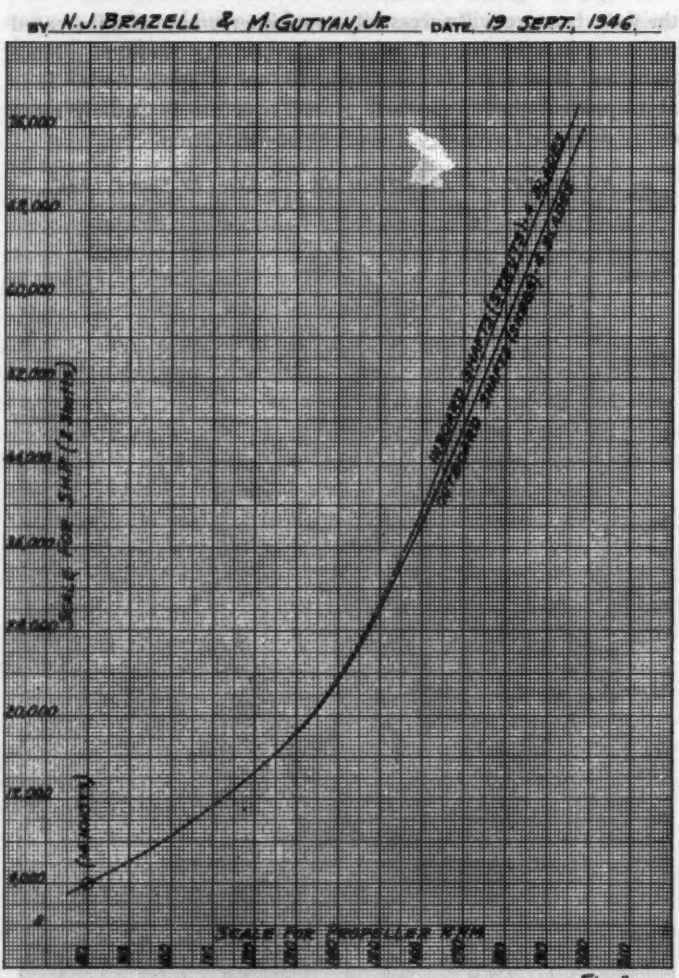


Fig. 4

Shaft Skegs.

The influence of skegs in reducing the rotation of the slip stream and the effect upon propeller performance at varying slip ratios is indicated in Figure 3. It should be noted that at the

cruising speeds the more efficient propellers, which are positioned behind the twin inboard skegs, are absorbing less horsepower than the strut supported outboard propellers. It is evident that the S.H.P. segregation is balanced at one shaft speed only, (188 turns). Twin inboard skegs afford good protection from underwater explosions and assist in drydocking the ship. Properly designed skegs and propellers should operate free from objectionable vibrations even in the high power installations. But there does not appear to be a clear cut advantage for the universal adoption of twin skegs disposed inboard on a four shaft propulsion installation.

When the twin skegs were disposed outboard with the strut supported shafts and propellers operating in a tunnel, the power segregation was reversed from what resulted with twin inboard skegs. (See Figure 4 and Figure 3). This outboard skeg arrangement offers promise. It should be possible to obtain any desired S.H.P. segregation throughout the entire speed range by altering the design of the tunnel (See Figure 5). With twin skegs outboard the maximum protection is afforded the propellers. For this specific application a set of new three bladed propellers was designed for the inboard shafts having a torque demand 7½ per cent less than was required for the original four bladed propellers, and hence equal power distribution was achieved at the full power condition. In addition, superior propulsive efficiency was achieved at the cruising speeds, with equal R.P.M.'s, because the more efficient skeg mounted propellers then absorbed a greater S.H.P. and developed more thrust than the companion strut mounted inboard propellers. The vessel, as built, had the conventional outboard turning propellers (See Scheme No. 1 in Figure 5). In order to verify the propriety of the tunnel design the rotation of the propellers was varied, (See Schemes No. 2 and No. 3) and the Model Basin reported the propulsive coefficients which are posted in Figure 5. It is clear from the self propelled model tests that the overall tunnel form was very good. However, an analysis of the standardization trials indicated a transverse velocity component in the induced slip stream to the inboard propellers which may explain the S.H.P. segregation paradox evidenced by Figure 3 and Figure 4.

Propulsive efficiency can be further improved on a propeller

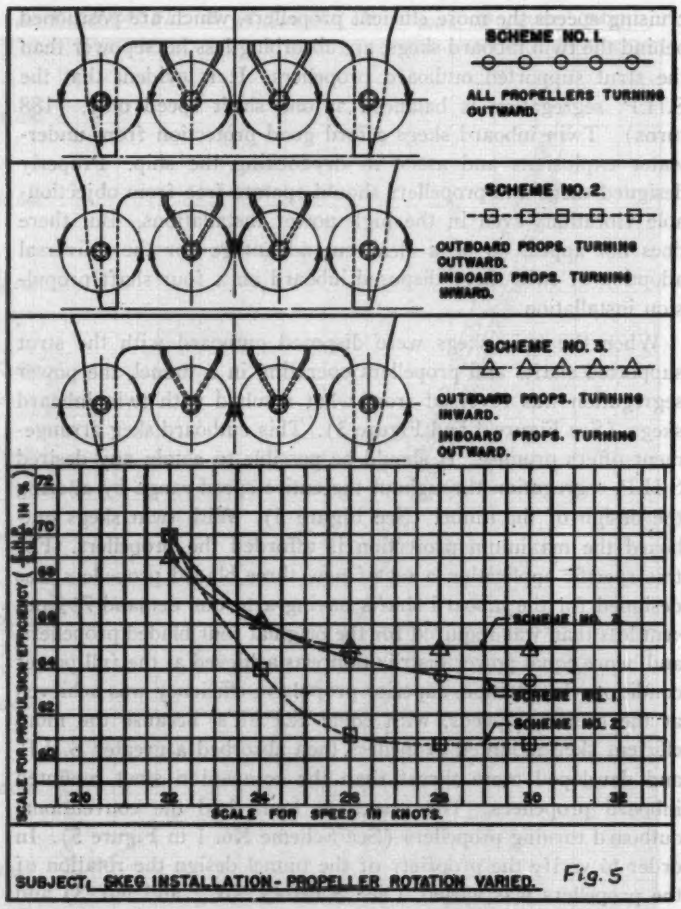


Fig. 5

operating behind a conventional skeg by providing a more symmetrical inflow to the propeller. In place of the conventional "V" sections, it is desirable to use full "U" or even bulbous sections in the skeg. The naval architect should note that what is thereby lost in added resistance per ton of displacement is doubly gained in an increased propeller efficiency. The molded lines and stern frame sections with a "U" shaped centerline skeg such as was used in the

single screw Navy fleet tugs illustrate a successful means of providing a more symmetrical flow to the propeller. The hull form of the fleet tugs produced a high propulsion efficiency under both the towing and free route operating conditions.

A promising arrangement for high powered ships of the future appears to be a quintuple screw drive with the centerline propeller positioned behind a skeg; the remaining four outboard shafts being carried by conventional double arm struts.

Propeller Blade Tip Clearance.

The most universal source of objectionable hull vibrations aboard ship is a consequence of insufficient propeller blade tip clearance at the upper arc of the propeller rotation. In low powered ships it is customary to provide a tip clearance of one inch per foot of propeller diameter for single screw ships, and for outboard strut supported propellers an allowance of two inches per foot of diameter is usually provided. This old one inch rule for the center propeller is satisfactory because of the athwartship symmetry of water flow at the propeller aperture and also because of the inherent structural stiffness of the stern framing. But this rule of thumb is not applicable for positioning the propellers of a modern ship in which over 50,000 shaft horsepower may be transmitted on a single shaft.

The variables which determine the intensity of the pressure field around a hydrofoil and the factors influencing the propeller blade tip clearance requirement are tabulated in Table I. A proposed tip clearance formula embodying the variables on Table I is shown on Table II. It will be noted that the proposed formula is a "natural", that is, no modifying coefficients are necessary for application to similar ships of varying sizes. The formula has produced well founded tip clearance dimensions for vessels varying from PT Boats to Battleships.

At first glance the tip clearance formula may appear to be too complex and the effort required to perform the calculation to be unwarranted. However, as we all know, it is now customary to require mechanics to pass a test of competence before qualifying to engage in certain types of skilled work. A comparable fitness test for the person who desires to position marine propellers is embodied in the mathematics contained in the tip clearance form-

TABLE I.

PROPELLER TIP CLEARANCE VARIABLES

Item	*Designation of Component*	*Symbol*	*Formula*	*Application for Increasing Component*
1	S.H.P. per Shaft.....	S.H.P.	P_1	P_1
2	Revolutions per Minute	R.P.M.	N	$\frac{1}{N}$
3	Torque.....	Q	$\frac{5252 \times P_1}{N}$	$\frac{5252 \times P_1}{N}$
4	Maximum Draft to Keel or Tip.....	d	d	$\frac{1}{d}$
5	Number of Blades.....	No. Bl.	No. Bl.	$\frac{1}{(\text{No. Bl.})}$
6	Propeller Diameter....	D	D	D^2
7	Propeller Pitch.....	P	P	P
8	Propeller Slip.....	S	%	(1+S)
9	Vena Contracta Influence.....	C	Sec (2×13°-30′)	Sec (2×13°-30′)
10	Speed Length Ratio...	$\frac{V}{\sqrt{L}}$	$\frac{V}{\sqrt{L}}$	$\frac{\sqrt{L}}{V \times 101.33}$
11	Stiffness Factor.....	K	$\frac{4EI\theta}{L}$	$\frac{L}{4 \times B \times d^2}$
12	Non-Axial Flow.....	N.A.F.	Sec $2\times\left\{\begin{matrix}\text{Rake}+\\ \text{Spread}+\\ \text{Squat}+\\ \text{Angle}\end{matrix}\right.$	$\frac{1}{\text{Sec}\,[2\times(R+S+S)]}$
13	Service Factor.....	Z	Z	Z

(B is the molded beam of the ship; L is the Length between perpendiculars, See Item No. 11.)

NOTE: The result of the Vena-Contracta or Non-Axial Flow influence is measured by the secant of twice the induced inflow angle (Sec 2θ), because the virtual increase of the propeller disc area is a factor in the required propeller tip clearance.

Mathematically: $D' = \frac{D}{\text{Cos}\,\theta} = D \times \text{Sec}\,\theta.$

The virtual disc area $= \frac{D^2}{\text{Cos}^2\theta}\left(\frac{\pi}{4}\right)$ or varies as $\frac{1}{\text{Cos}^2\theta}.$

But $\text{Cos}^2\theta = \frac{1 + \text{Cos}\,2\theta}{2}.$

Since there also exists a velocity component of rotation in the slip stream which constitutes a virtual non-axial flow with an effect that is comparable to the fore and aft convergence outlined by the tip vortices (which may be observed in the Hydraulic tunnel) the total vena-contracta influence is: $\frac{1}{\text{Cos}^2\theta} \times \frac{1}{\text{Cos}^2\theta} = \frac{1}{\left(\frac{1 + \text{Cos}\,2\theta}{2}\right)^2}$ which, for small angles, is equivalent to secant 2θ.

PROPELLER BLADE TIP CLEARANCE

COMPUTED VALUE BY BRAZELL'S FORMULAE VERSUS ACTUAL OR DESIGNED

$$T.C. = Z \times \sqrt[3]{\frac{15}{\text{SHP} \times S.H.P \times G^3 \times P \times (1 - S)\% \times \sqrt{V} \times [\sec E \times (180 - 30^\circ)]^3}{R \times NO.BLADES \times D \times G^4 \times V \times [\sec E \times (90 + S + S_2 \text{ ANGLE})]^3}} \times \sqrt[3]{\left[\frac{L^{0.5} \times L^{0.5} \times L^{0.5} \times L^{1.0}}{L^{1.5} \times L^{0.5} \times L^{0.5} \times L^{0.5}}\right]^{\frac{1}{3}}} \times \sqrt[3]{\left(\frac{L}{G^2}\right)^{\frac{1}{3}}} \times L$$

T.C. IS THE TIP CLEARANCE IN FEET AT TOPSIDE REQUIRED FOR ACCEPTABLE VERTICAL VIBRATIONS.
SHP IS THE S.H.P. PER SHAFT
G IS THE GRAM OF SHIP (DISPLACED)
G IS THE DIAMETER IN FEET
d IS THE DRAFT, MAX. TO BASE LINE LOADED OR TIP BELOW BASE. d = C B + 2.33 FOR CARGO SHIPS.
L IS THE LENGTH OF SHIP (L.W.L.)
R IS THE R.P.M.
P IS THE PITCH IN FEET
A IS THE SHAFT RAKE ANGLE
S IS THE TRUE SLIP IN %
S_1 IS THE SHAFT SPREAD ANGLE
S_2 IS THE SKEW ANGLE
V IS THE SHIP SPEED IN KNOTS (LESS ½ % FOUL)
SEC E X (180-30°) IS THE VERNA-CONTRACTA FACTOR
Z IS THE FACTOR OF SERVICE Z = 1.00 FOR COMBATANT AND PASSENGER SHIPS Z = 0.75 FOR DESTROYERS AND TRANSPORTS
Z = 0.60 FOR TWIN SCREW MTB CRAFT AND AUXILIARIES Z = 0.40 FOR SINGLE SCREW CARGO SHIPS AND TUGS
STIFFNESS FACTOR = 4 E 30 ÷ L (T.C. APPLICATION = L ÷ 4 X 10⁶, OR L ÷ 4 X 30 X d^3)
MINIMUM FORE AND AFT CLEARANCE FROM LEADING EDGE IN APERTURE OR TO STRUT ARM = T.C. × .05 D (50% LESS FOR GUIDE VANE INSTALLATIONS.)

SHIP DESIGNATION	NO. PROPS	TYPE OF APPENDAGE	TIP CLEARANCE CALCULATION IN FEET	TIP CLEARANCE COMPUTED	TIP CLEARANCE ACTUAL OR DESIGNED	REMARKS
C1-S-D1	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 1130 \times 14.50 \times 14.50 \times 14.500 \times 1.348 \times 360 \times 19.71 \times 1.4}{60 \times 4 \times 64 \times 18.5 \times 18.5 \times 18.5 \times 18.5 \times 18.5 \times 10.3 \times 1.00}} = 0.4 \times \sqrt[3]{86.43}$	14 1/2"	24"	
SI-M-AV1	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 1810 \times 11.00 \times 11.00 \times 7.930 \times 1.693 \times 360 \times 17.59 \times 1.4}{150 \times 5 \times 50 \times 18.0 \times 18.0 \times 18.0 \times 18.0 \times 18.0 \times 1.00}} = 0.4 \times \sqrt[3]{7.763}$	9 1/2"	42"	
S2 STEAM	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 6000 \times 19.00 \times 19.00 \times 19.00 \times 1.375 \times 1.38 \times 438.5 \times 20.94 \times 1.4}{92 \times 4 \times 63 \times 25.75 \times 25.75 \times 25.75 \times 25.75 \times 25.75 \times 1.00}} = 0.4 \times \sqrt[3]{51.264}$	18 3/8"	19"	
C2 DIESEL	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 5500 \times 17.33 \times 17.33 \times 18.875 \times 1.40 \times 438.5 \times 20.94 \times 1.4}{92 \times 4 \times 63 \times 25.75 \times 25.75 \times 25.75 \times 25.75 \times 25.75 \times 1.00}} = 0.4 \times \sqrt[3]{53.60}$	18 1/4"	28"	
VC2-M-AP4	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 5860 \times 14.00 \times 14.00 \times 11.80 \times 1.471 \times 443 \times 21.10 \times 1.4}{160 \times 4 \times 62 \times 28.5 \times 28.5 \times 28.5 \times 28.5 \times 28.5 \times 1.00}} = 0.4 \times \sqrt[3]{9.66}$	10 7/16"	66"	
VC2-S-AP1	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 4850 \times 19.00 \times 19.00 \times 19.00 \times 1.354 \times 443 \times 21.10 \times 1.4}{88 \times 4 \times 62 \times 29.5 \times 29.5 \times 29.5 \times 29.5 \times 29.5 \times 1.00}} = 0.4 \times \sqrt[3]{48.14}$	17 3/8"	36"	
VC2-S-AP2	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 6000 \times 18.25 \times 18.25 \times 17.500 \times 1.348 \times 443 \times 21.10 \times 1.4}{100 \times 4 \times 62 \times 25.5 \times 25.5 \times 25.5 \times 25.5 \times 25.5 \times 1.00}} = 0.4 \times \sqrt[3]{59.37}$	18 1/2"	48"	
VC2-S-AP3 (SISTER SHIP)	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 6500 \times 20.50 \times 20.50 \times 22.500 \times 1.338 \times 443 \times 21.10 \times 1.4}{95 \times 4 \times 62 \times 28.0 \times 28.0 \times 28.0 \times 28.0 \times 28.0 \times 1.00}} = 0.4 \times \sqrt[3]{90.16}$	22"	21 1/2"	
C3	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 6900 \times 21.67 \times 21.67 \times 21.670 \times 1.290 \times 469.5 \times 21.67 \times 1.4}{88 \times 4 \times 69.5 \times 27.25 \times 27.25 \times 27.25 \times 27.25 \times 17.0 \times 1.00}} = 0.4 \times \sqrt[3]{88.87}$	21 5/8"	17 1/2"	
C4-S-B1	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 9000 \times 19.00 \times 19.00 \times 18.278 \times 1.348 \times 500.5 \times 22.37 \times 1.4}{104 \times 4 \times 71.5 \times 30.0 \times 30.0 \times 30.0 \times 30.0 \times 17.4 \times 1.006}} = 0.4 \times \sqrt[3]{38.78}$	16 5/8"	40 3/8"	
C4-S-A1	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 9000 \times 21.67 \times 21.67 \times 21.670 \times 1.290 \times 500.5 \times 22.37 \times 1.4}{87 \times 4 \times 71.5 \times 30.0 \times 30.0 \times 30.0 \times 30.0 \times 18.4 \times 1.00}} = 0.4 \times \sqrt[3]{63.468}$	19 1/8"	21"	
T1-S-A1	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 500 \times 6.52 \times 6.52 \times 4.500 \times 1.498 \times 212.5 \times 14.58 \times 1.4}{300 \times 3 \times 37 \times 13.0 \times 13.0 \times 13.0 \times 13.0 \times 10.4 \times 1.00}} = 0.4 \times \sqrt[3]{1.82}$	8 1/16"	18"	
T1-S-A1	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 500 \times 7.00 \times 7.00 \times 4.500 \times 1.498 \times 212.5 \times 14.58 \times 1.4}{290 \times 4 \times 37 \times 13.0 \times 13.0 \times 13.0 \times 13.0 \times 10.6 \times 1.00}} = 0.4 \times \sqrt[3]{1.257}$	8 1/8"	16 1/8"	
B3-S-A1	1	SKEG	$0.4 \times \sqrt[3]{\frac{13 \times 1130 \times 13.50 \times 13.50 \times 16.875 \times 1.375 \times 438.5 \times 19.50 \times 1.4}{80 \times 4 \times 41.6 \times 17.5 \times 17.5 \times 17.5 \times 17.5 \times 11.8 \times 1.00}} = 0.4 \times \sqrt[3]{23.42}$	13 5/8"	16 3/8"	
P1-S2-L2	2	STRUTS	$0.75 \times \sqrt[3]{\frac{15 \times 7000 \times 12.33 \times 12.33 \times 12.670 \times 1.160 \times 400 \times 20.00 \times 1.4}{100 \times 4 \times 86 \times 17.6 \times 17.6 \times 17.6 \times 17.6 \times 17.6 \times 19.4 \times 1.00}} = 0.75 \times \sqrt[3]{19.82}$	23 1/2"	24"	
P2-S2-R2	2	SHAFT BOSS & STRUTS	$0.75 \times \sqrt[3]{\frac{13 \times 6900 \times 18.00 \times 18.00 \times 22.700 \times 1.290 \times 390 \times 4.29 \times 1.4}{100 \times 4 \times 79.5 \times 29.0 \times 29.0 \times 29.0 \times 29.0 \times 29.0 \times 1.00}} = 0.75 \times \sqrt[3]{61.8}$	39"	36 3/8"	

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71-M-41	1	SKEG	0.4 X $\sqrt[5]{\text{[illegible]}}$ = 0.4 X $\sqrt[5]{1.287}$	6 3/4"	16 11/16"	
83-S-A1	1	SKEG	0.4 X $\sqrt[5]{\text{[illegible]}}$ = 0.4 X $\sqrt[5]{23.42}$	13 1/4"	16 11/16"	
P1-S2-L2	2	STRUTS	0.75 X $\sqrt[5]{\text{[illegible]}}$ = 0.75 X $\sqrt[5]{19.88}$	23 1/4"	24"	
P2-S2-R2	2	SHAFT BOSS & STRUTS	0.75 X $\sqrt[5]{\text{[illegible]}}$ = 0.75 X $\sqrt[5]{[illegible]}$	39"	34 11/16"	
P2-SE2-R1 (NO RAKE)	2	STRUTS	0.75 X $\sqrt[5]{\text{[illegible]}}$ = 0.75 X $\sqrt[5]{[illegible]}$	39 1/8"	40"	
P2-SE2-R1 (NO RAKE)	2	STRUTS	0.75 X $\sqrt[5]{\text{[illegible]}}$ = 0.75 X $\sqrt[5]{85.0887}$	39 1/8"		
S4-SE2-R01 (ARA 32)	2	STRUTS	0.6 X $\sqrt[5]{\text{[illegible]}}$ = 0.6 X $\sqrt[5]{24.40}$	20 1/8"	19 1/8"	LONGITUDINAL VIBRATION=10 MILLS DBL AMP AT 105 R.P.M. ON PORT SHAFT @ 16 MILLS AT 106 R.P.M. ON STBD SHAFT.
S4-SE2-S83	2	BOSSING	0.75 X $\sqrt[5]{\text{[illegible]}}$ = 0.75 X $\sqrt[5]{20.15}$	24 3/4"	23 11/16"	
EC2-S1 LIBERTY SHIP	1	SKEG	0.4 X $\sqrt[5]{\text{[illegible]}}$ = 0.4 X $\sqrt[5]{22.8339}$	13 1/8"	6"	
PROPELLER	4	BOSSING	1.0 X $\sqrt[5]{\text{[illegible]}}$ = 1.0 X $\sqrt[5]{31.928}$	37 3/8"	79 1/2"	
ORIZABA (3 BLADES)	2	BOSSING	1.0 X $\sqrt[5]{\text{[illegible]}}$ = 1.0 X $\sqrt[5]{24.81}$	35"	25 1/2"	
ORIZABA (5 BLADES)	2	BOSSING	1.0 X $\sqrt[5]{\text{[illegible]}}$ = 1.0 X $\sqrt[5]{11.6165}$	27 5/16"	32 1/2"	
10000 TON TROOP TRANSPORT	2	STRUT	.75 X $\sqrt[5]{\text{[illegible]}}$ = .75 X $\sqrt[5]{21.3788}$	24.975"	28.92"	
JAPANESE FUEL SHIP "HAMOI"	2	STRUT	.60 X $\sqrt[5]{\text{[illegible]}}$ = .60 X $\sqrt[5]{16.7169}$	18.615"	13.692"	
S.S. BRANDTON	2	STRUT	.60 X $\sqrt[5]{\text{[illegible]}}$ = .60 X $\sqrt[5]{2.8931}$	10.26"	13.632"	
S.S. REPUBLIC	2	BOSSES	.75 X $\sqrt[5]{\text{[illegible]}}$ = .75 X $\sqrt[5]{26.6967}$	26.976"	6.60"	
MATSON LINER	2	BOSSES	.75 X $\sqrt[5]{\text{[illegible]}}$ = .75 X $\sqrt[5]{82.231}$	29.141"	28.45"	
TWIN SCREW TANKER	2	BOSSES	.60 X $\sqrt[5]{\text{[illegible]}}$ = .60 X $\sqrt[5]{4.9852}$	12.996"	14.16"	
S.S. ALAMAR	2	BOSSES	.60 X $\sqrt[5]{\text{[illegible]}}$ = .60 X $\sqrt[5]{4.3260}$	11.729"	3.0"	
HAWAIIAN PACIFIC LINER	2	BOSSES	1.0 X $\sqrt[5]{\text{[illegible]}}$ = 1.0 X $\sqrt[5]{40.394}$	41.178"	31.8"	
CLYDE LINE STEAMER	2	STRUTS	1.0 X $\sqrt[5]{\text{[illegible]}}$ = 1.0 X $\sqrt[5]{28.7885}$	36.768"	32.4"	
TWIN SCREW MOTOR VESSEL	2	BOSSES	.80 X $\sqrt[5]{\text{[illegible]}}$ = .80 X $\sqrt[5]{12.0205}$	16.496"	18.0"	
TWIN SCREW STEAMER MATSON LINE	2	BOSSES	.80 X $\sqrt[5]{\text{[illegible]}}$ = .80 X $\sqrt[5]{13.3173}$	17.264"	7.44"	
PRESIDENT MONROE & PRESIDENT WASHINGTON	2	STRUTS	1.0 X $\sqrt[5]{\text{[illegible]}}$ = 1.0 X $\sqrt[5]{100.6764}$	58.896"	29.88"	
FREIGHT & PASSENGER SHIP	2	BOSSES	.75 X $\sqrt[5]{\text{[illegible]}}$ = .75 X $\sqrt[5]{65.2872}$	34.34"	21.876"	
CARGO & PASSENGER SHIP	2	BOSSES	.75 X $\sqrt[5]{\text{[illegible]}}$ = .75 X $\sqrt[5]{14.5101}$	21.982"	20.226"	
CARGO & PASSENGER	2	BOSSES	.75 X $\sqrt[5]{\text{[illegible]}}$ = .75 X $\sqrt[5]{23.6849}$	26.88"	28.90"	

T1-M-A1	1	SKEG	0.4 x $\sqrt[3]{15 \times 800 \times 6.82 \times 6.82 \times 4.801 \times 1.495 \times 212.5 \times 14.88 \times 1.4 \over 800 \times 3 \times 27 \times 13.0 \times 13.0 \times 13.0 \times 13.0 \times 10.4 \times 1.00}$ = 0.4 x $\sqrt[3]{1.02}$	8 $\frac{5}{16}$"	10"	
T1-M-A1	1	SKEG	0.4 x $\sqrt[3]{13 \times 800 \times 7.00 \times 7.00 \times 4.900 \times 1.490 \times 212.5 \times 14.88 \times 1.4 \over 280 \times 4 \times 27 \times 13.0 \times 13.0 \times 13.0 \times 13.0 \times 10.6 \times 1.00}$ = 0.4 x $\sqrt[3]{1.287}$	9 $\frac{1}{16}$"	10 $\frac{3}{16}$"	
N3-S-A1	1	SKEG	0.4 x $\sqrt[3]{13 \times 1130 \times 13.80 \times 13.80 \times 10.975 \times 1.378 \times 294.8 \times 19.06 \times 1.4 \over 80 \times 4 \times 41.8 \times 17.5 \times 17.5 \times 17.5 \times 17.5 \times 11.8 \times 1.00}$ = 0.4 x $\sqrt[3]{23.42}$	13 $\frac{5}{8}$"	16 $\frac{3}{16}$"	
P1-S2-L2	2	STRUTS	0.75 x $\sqrt[3]{12 \times 7000 \times 12.33 \times 12.33 \times 12.870 \times 1.360 \times 400 \times 20.00 \times 1.4 \over 160 \times 4 \times 56 \times 17.6 \times 17.6 \times 17.6 \times 17.6 \times 10.4 \times 1.00}$ = 0.75 x $\sqrt[3]{19.82}$	23 $\frac{1}{4}$"	26"	
P1-S2-R2	2	SHAFT BOSS & STRUTS	0.75 x $\sqrt[3]{13 \times 8500 \times 18.00 \times 18.00 \times 12.000 \times 1.500 \times 540 \times 22.4 \times 1.4 \over 400 \times 4 \times 78.9 \times 25.0 \times 25.0 \times 25.0 \times 25.0 \times 20.8 \times 1.00}$ = 0.75 x $\sqrt[3]{.6828}$	39"	34 $\frac{3}{16}$"	
P2-SE2-R1 (No. RANK)	2	STRUTS	0.75 x $\sqrt[3]{13 \times 9000 \times 18.66 \times 18.66 \times 12.44 \times 1.21 \times 590 \times 24.30 \times 1.4 \over 180 \times 3 \times 75.5 \times 25.0 \times 25.0 \times 25.0 \times 25.0 \times 20.8 \times 1.000}$ = 0.75 x $\sqrt[3]{.6.38676}$	39 $\frac{3}{8}$"	46"	
P2-SE2-R1 (NO RANK)	2	STRUTS	0.75 x $\sqrt[3]{13 \times 9000 \times 18.5 \times 18.5 \times 19.25 \times 1.21 \times 590 \times 24.30 \times 1.4 \over 120 \times 3 \times 75.5 \times 25.0 \times 25.0 \times 25.0 \times 25.0 \times 20.6 \times 1.000}$ = 0.75 x $\sqrt[3]{89.0837}$	39 $\frac{3}{8}$"		
C4-S2E2-R01 (ARA S2)	2	STRUTS	0.6 x $\sqrt[3]{13 \times 7000 \times 11.50 \times 11.50 \times 12.885 \times 1.210 \times 400 \times 20.00 \times 1.4 \over 130 \times 4 \times 56.7 \times 16.5 \times 16.5 \times 16.5 \times 16.5 \times 17.8 \times 1.003}$ = 0.6 x $\sqrt[3]{24.40}$	20 $\frac{7}{8}$"	19 $\frac{3}{4}$"	Lower Forward, Weight Fwd = 19 Units Each Aft 165 Units, On Port Shaft @ 14 Units Aft 156 Units on Stbd Shaft.
C4-S2-R03	2	BOSSING	0.75 x $\sqrt[3]{13 \times 4140 \times 12.25 \times 12.25 \times 13.710 \times 1.221 \times 490 \times 22.14 \times 1.4 \over 160 \times 4 \times 65 \times 16.9 \times 16.9 \times 16.9 \times 16.9 \times 15.2 \times 1.00}$ = 0.75 x $\sqrt[3]{20.16}$	24 $\frac{7}{8}$"	20 $\frac{3}{16}$"	
EC2-S2-R1 LIBERTY SHIP	1	SKEGS	0.4 x $\sqrt[3]{13 \times (2750 \times 38) \times 18.5 \times 18.5 \times 14.5 \times 1.24 \times 447.3 \times 21.5 \times 1.4 \over 78 \times 4 \times 54.6 \times 21.05 \times 21.05 \times 21.05 \times 21.05 \times 11.0 \times 1.00}$ = 0.4 x $\sqrt[3]{22.0329}$	13 $\frac{5}{8}$"	6"	
PERFORMANCE	4	BOSSING	1.0 x $\sqrt[3]{13 \times 6000 \times 15.84 \times 15.84 \times 47.000 \times 1.210 \times 849 \times 31.01 \times 1.4 \over 194 \times 4 \times 17.8 \times 34.4 \times 34.4 \times 34.4 \times 34.4 \times 30.2 \times 1.00}$ = 1.0 x $\sqrt[3]{31.522}$	37 $\frac{7}{8}$"	79 $\frac{1}{4}$"	

ORIZABA (3 BLADES)	2	BOSSING	1.0 x $\sqrt[3]{13 \times 4125 \times 15.50 \times 15.50 \times 16.750 \times 1.230 \times 474.5 \times 20.64 \times 1.4 \over 120 \times 3 \times 60 \times 24.5 \times 24.5 \times 24.5 \times 24.5 \times 17.0 \times 1.00}$ = 1.0 x $\sqrt[3]{24.81}$	38"	25 $\frac{1}{4}$"	
ORIZABA (5 BLADES)	2	BOSSING	1.0 x $\sqrt[3]{13 \times 4125 \times 14.33 \times 14.33 \times 15.833 \times 1.200 \times 434.5 \times 20.64 \times 1.4 \over 170 \times 5 \times 60.0 \times 24.50 \times 24.50 \times 24.50 \times 24.50 \times 17.0 \times 1.00}$ = 1.0 x $\sqrt[3]{14.6163}$	27 $\frac{3}{16}$"	32 $\frac{1}{4}$"	
10000 TON TROOP TRANSPORT	2	STRUT	.75 x $\sqrt[3]{13 \times 4000 \times 17.5 \times 17.5 \times 18.9 \times (10.246) \times 595 \times 28.19 \times 1.4 \over 100 \times 3 \times 72.2 \times 30 \times 30 \times 30 \times 30 \times 16 \times 1.0}$ = .75 x $\sqrt[3]{21.5765}$	24.975"	28.82"	
JAPANESE FUEL SHIP "NAMOI"	2	STRUT	.60 x $\sqrt[3]{13 \times 4320 \times 16.083 \times 16.083 \times 16.975 \times (10.226) \times 6960 \times 25.27 \times 1.4 \over 123 \times 3 \times 67.0 \times 20.21 \times 20.21 \times 20.21 \times 20.21 \times 15.25 \times 1.0}$ = .60 x $\sqrt[3]{16.7159}$	18.410"	13.632"	
S.S. MISSISSIPPI	2	STRUT	.60 x $\sqrt[3]{13 \times 5200 \times 11.964 \times 11.964 \times 8.531 \times (10.346) \times 341 \times 16.46 \times 1.4 \over 154 \times 3 \times 48.0 \times 22.33 \times 22.33 \times 22.33 \times 22.33 \times 10.9 \times 1.0}$ = .60 x $\sqrt[3]{2.8981}$	10.25"	12.682"	
S.S. REPUBLIC	2	BOSSES	.75 x $\sqrt[3]{13 \times 3050 \times 17.5 \times 17.5 \times 19.75 \times (10.299) \times 595 \times 24.48 \times 1.4 \over 79.5 \times 3 \times 68.5 \times 32.9 \times 32.9 \times 32.9 \times 32.9 \times 13.5 \times 1.0}$ = .75 x $\sqrt[3]{26.6367}$	24.874"	6.60"	
MATSON LINER	2	BOSSES	.75 x $\sqrt[3]{13 \times 10575 \times 19 \times 19 \times 20.5 \times (10.138) \times 577 \times 24.02 \times 1.4 \over 146 \times 3 \times 83 \times 26 \times 26 \times 26 \times 26 \times 21.0 \times 1.0}$ = .75 x $\sqrt[3]{82.251}$	39.141"	38.40"	
TWIN SCREW TANKER	2	BOSSES	.60 x $\sqrt[3]{13 \times 1520 \times 14 \times 14 \times 8.35 \times (10.392) \times 472.25 \times 21.73 \times 1.4 \over 110 \times 4 \times 65 \times 27 \times 27 \times 27 \times 27 \times 11.5 \times 1.0}$ = .60 x $\sqrt[3]{4.9852}$	12.395"	14.16"	
S.S. ALASKAN	2	BOSSES	.60 x $\sqrt[3]{13 \times 7245 \times 13.036 \times 13.036 \times 13.036 \times 13.036 \times (10.260) \times 669 \times 25.86 \times 1.4 \over 116 \times 4 \times 57 \times 31.854 \times 31.854 \times 31.854 \times 31.854 \times 13 \times 1.0}$ = .60 x $\sqrt[3]{4.3260}$	11.729"	3.0"	
MANHATTA PACIFIC LINER	2	BOSSES	1.0 x $\sqrt[3]{13 \times 9625 \times 18.5 \times 18.5 \times 17.29 \times (10.264) \times 590 \times 24.59 \times 1.4 \over 122 \times 3 \times 80 \times 30 \times 30 \times 30 \times 30 \times 19 \times 1.0}$ = 1.0 x $\sqrt[3]{40.394}$	41.172"	31.0"	
CLYDE LINE STEAMER	2	STRUTS	1.0 x $\sqrt[3]{13 \times 4395 \times 13.34 \times 13.34 \times 14.41 \times (10.208) \times 468 \times 24.3 \times 1.4 \over 119 \times 3 \times 62 \times 19 \times 19 \times 19 \times 19 \times 19 \times 1.0}$ = 1.0 x $\sqrt[3]{28.7685}$	34.766"	32.4"	
TWIN SCREW MOTOR VESSEL	2	BOSSES	.60 x $\sqrt[3]{13 \times 2700 \times 16 \times 16 \times 16.48 \times (10.268) \times 522 \times 22.86 \times 1.4 \over 90.4 \times 3 \times 70 \times 29.5 \times 29.5 \times 29.5 \times 29.5 \times 13.5 \times 1.0}$ = .60 x $\sqrt[3]{12.0205}$	18.495"	18.0"	
MOORE-McCORMACK LINE STEAMSHIP	2	BOSSES	.60 x $\sqrt[3]{13 \times 1066 \times 11 \times 11 \times 14.2 \times (10.234) \times 310 \times 17.61 \times 1.4 \over 129.5 \times 3 \times 48 \times 16.75 \times 16.75 \times 16.75 \times 16.75 \times 15 \times 1.0}$ = .60 x $\sqrt[3]{13.3173}$	17.064"	7.44"	
ALEXANDRIA-MOUNT & MT. VERNON	2	BOSSES	1.0 x $\sqrt[3]{13 \times 20675 \times 19.86 \times 19.86 \times 22.46 \times (10.228) \times 690 \times 24.09 \times 1.4 \over 119.5 \times 3 \times 72 \times 32.56 \times 32.56 \times 32.56 \times 32.56 \times 24.0 \times 1.0}$ = 1.0 x $\sqrt[3]{100.6764}$	58.536"	29.88"	
FREIGHT & PASSENGER SHIP	2	BOSSES	.75 x $\sqrt[3]{13 \times 7925 \times 18 \times 18 \times 19.5 \times (10.185) \times 625 \times 25 \times 1.4 \over 113.7 \times 3 \times 77.5 \times 24.5 \times 24.5 \times 24.5 \times 24.5 \times 20 \times 1.0}$ = .75 x $\sqrt[3]{65.2872}$	36.34"	21.876"	
CARGO & PASSENGER SHIP	2	BOSSES	.75 x $\sqrt[3]{13 \times 5250 \times 18.75 \times 18.75 \times 18.75 \times (10.264) \times 480 \times 21.91 \times 1.4 \over 83.9 \times 3 \times 61.5 \times 26 \times 26 \times 26 \times 26 \times 13 \times 1.0}$ = .75 x $\sqrt[3]{14.5181}$	21.952"	20.256"	
CARGO & PASSENGER SHIP	2	BOSSES	.75 x $\sqrt[3]{13 \times 6925 \times 18 \times 18 \times 17.75 \times (10.211) \times 607 \times 24.64 \times 1.4 \over 115 \times 3 \times 80 \times 32 \times 32 \times 32 \times 32 \times 18 \times 1.0}$ = .75 x $\sqrt[3]{25.6843}$	26.58"	23.90"	
TWIN SCREW CARGO SHIP	2	BOSSES	.60 x $\sqrt[3]{13 \times 2970 \times 17.5 \times 17.5 \times 18.54 \times (10.238) \times 470 \times 21.66 \times 1.4 \over 73.1 \times 4 \times 63 \times 30 \times 30 \times 30 \times 30 \times 14 \times 1.0}$ = .60 x $\sqrt[3]{13.77307}$	17.256"	19.80"	
TWIN SCREW PASSENGER STEAMER	2	BOSSES	1.0 x $\sqrt[3]{13 \times 11300 \times 18.5 \times 18.5 \times 18.5 \times (10.189) \times 650 \times 28.10 \times 1.4 \over 139.3 \times 3 \times 81 \times 30.25 \times 30.25 \times 30.25 \times 30.25 \times 20.0 \times 1.0}$ = 1.0 x $\sqrt[3]{45.1195}$	42.72"	25.80"	
SANTA						

[illegible]	[illegible]	[illegible]	$\sqrt[3]{\frac{\ldots}{\ldots \times 3 \times 61.5 \times 26 \times 26 \times 26 \times 26 \times 15 \times 1.0}}$	[illegible]	[illegible]	[illegible]	
CARGO & PASSENGER SHIP	2	BOSSES	$.75 \times \sqrt[3]{\frac{13 \times 6925 \times 18 \times 18 \times 17.75 \times (1+.211) \times 607 \times 24.64 \times 1.4}{113 \times 3 \times 80 \times 32 \times 32 \times 32 \times 32 \times 18 \times 1.0}}$	$= .75 \times \sqrt[3]{25.6843}$	24.88"	33.90"	
TWIN SCREW CARGO SHIP	2	BOSSES	$.60 \times \sqrt[3]{\frac{13 \times 7070 \times 17.5 \times 17.5 \times 19.5 \times (1+.\ldots) \times 470 \times \ldots \times 1.4}{73.1 \times 4 \times 63 \times 30 \times 30 \times 30 \times 30 \times 14 \times 1.0}}$	$= .60 \times \sqrt[3]{13.77307}$	17.258"	19.80"	
TWIN SCREW PASSENGER STEAMER	2	BOSSES	$1.0 \times \sqrt[3]{\frac{13 \times 11300 \times 18.5 \times 18.5 \times 19 \times (1+.189) \times 630 \times 25.10 \times 1.4}{129.3 \times 3 \times 81 \times 30.25 \times 30.25 \times 30.25 \times 30.25 \times 20.0 \times 1.0}}$	$= 1.0 \times \sqrt[3]{45.1195}$	42.72"	25.80"	
SANTA CLARA	2	BOSSES	$.75 \times \sqrt[3]{\frac{13 \times 4850 \times 16.5 \times 16.5 \times 17.5 \times (1+.209) \times 498.9 \times 22.34 \times 1.4}{113.5 \times 3 \times 63.75 \times 25.395 \times 25.395 \times 25.395 \times 25.395 \times 18 \times 1.0}}$	$= .75 \times \sqrt[3]{34.8722}$	29.40"	28.80"	
TWIN SCREW PASSENGER & CARGO STEAMER	2	BOSSES	$.75 \times \sqrt[3]{\frac{13 \times 2795 \times 16.75 \times 16.75 \times 16.75 \times (1+.24) \times 517 \times 22.74 \times 1.4}{92.5 \times 3 \times 68 \times 28 \times 28 \times 28 \times 28 \times 15 \times 1.0}}$	$= .75 \times \sqrt[3]{20.9554}$	24.81"	21.96	
S.S MANHATTAN	2	BOSSES	$1.0 \times \sqrt[3]{\frac{13 \times 11375 \times 19 \times 19 \times 20 \times (1+.227) \times 685 \times 26.17 \times 1.4}{124 \times 4 \times 86 \times 29 \times 29 \times 29 \times 29 \times 20 \times 1.0}}$	$= 1.0 \times \sqrt[3]{45.6061}$	42.876"	24.30"	
444'-2" OA VESSEL	2	BOSSES	$1.0 \times \sqrt[3]{\frac{13 \times 2415 \times 15.25 \times 15.25 \times 17 \times (1+.162) \times 430 \times 20.74 \times 1.4}{101.1 \times 3 \times 60 \times 24.188 \times 24.188 \times 24.188 \times 24.188 \times 16 \times 1.0}}$	$= 1.0 \times \sqrt[3]{18.0684}$	31.488	37.90"	
PASSENGER & CARGO VESSEL	2	BOSSES	$.75 \times \sqrt[3]{\frac{13 \times 5520 \times 16.5 \times 16.5 \times 17 \times (1+.222) \times 500 \times 22.36 \times 1.4}{114.2 \times 3 \times 70 \times 25 \times 25 \times 25 \times 25 \times 19 \times 1.0}}$	$= .75 \times \sqrt[3]{35.6896}$	29.63	35.40"	
PASSENGER & FREIGHT SHIPS EASTERN S.S. LINES	2	STRUTS & SPADE RUDDER	$.60 \times \sqrt[3]{\frac{13 \times 3153 \times 12.75 \times 12.75 \times 13.75 \times (1+.213) \times 400.25 \times 20.01 \times 1.4}{148.9 \times 3 \times 61 \times 18 \times 18 \times 18 \times 18 \times 18 \times 1.0}}$	$= .60 \times \sqrt[3]{24.1985}$	20.82"	18.0"	
SCOTT PASSENGER AND FREIGHT SHIP	2	BOSSES	$.60 \times \sqrt[3]{\frac{13 \times 5280 \times 19 \times 19 \times 21.64 \times (1+.181) \times 494.75 \times 22.24 \times 1.4}{97.1 \times 3 \times 69.667 \times 26 \times 26 \times 26 \times 26 \times 19 \times 1.0}}$	$= .6 \times \sqrt[3]{55.0495}$	27.39"	15.72"	
SANTA BARBARA	2	BOSSES	$.75 \times \sqrt[3]{\frac{13 \times 4595 \times 15.25 \times 15.25 \times 18.75 \times (1+.288) \times 480 \times 21.91 \times 1.4}{110 \times 4 \times 63.75 \times 24.813 \times 24.813 \times 24.813 \times 24.813 \times 18 \times 1.0}}$	$= .75 \times \sqrt[3]{25.8092}$	26.60"	21.0"	
PANAMA MAIL STEAMERS SANTA ROSA TYPE	2	BOSSES	$1.0 \times \sqrt[3]{\frac{13 \times 5110 \times 17.5 \times 17.5 \times 19.5 \times (1+.212) \times 500 \times 22.36 \times 1.4}{94.9 \times 4 \times 72 \times 25.2 \times 25.2 \times 25.2 \times 25.2 \times 18.5 \times 1.0}}$	$= 1.0 \times \sqrt[3]{36.9075}$	39.948"	45.24"	
TWIN SCREW PASSENGER & CARGO MOTOR-SHIP	2	BOSSES	$.75 \times \sqrt[3]{\frac{13 \times 4140 \times 15.0 \times 15.0 \times 18.5 \times (1+.327) \times 459 \times 21.42 \times 1.4}{110 \times 3 \times 61.5 \times 26 \times 26 \times 26 \times 26 \times 17 \times 1.0}}$	$= .75 \times \sqrt[3]{25.9536}$	26.68"	45.0"	
1942 PT	3	STRUT	$1.0 \times \sqrt[3]{\frac{13 \times 1350 \times 2.33 \times 2.33 \times 2.33 \times 1.23 \times 73.6 \times 3.573 \times 1.40}{2400 \times 3 \times 17.5 \times 5.75 \times 5.75 \times 5.75 \times 5.75 \times 42 \times (\sec 2 \ldots)}}$	$= 1.0 \times \sqrt[3]{.0271}$	3 1/8"	3 5/32"	
[illegible]	2	STRUT	$0.75 \times \sqrt[3]{\frac{[illegible]}{[illegible]}}$	$0.75 \times \sqrt[3]{54.50}$	34.157"	25"*	
[illegible] LÜTZOW	CENTER LINE	BOSS [illegible]	$.80 \times \sqrt[3]{\frac{[illegible]}{[illegible]}}$	$.80 \times \sqrt[3]{29.92}$	29.80"	23.52"	
	TWO OUTBD	STRUT	$1.0 \times \sqrt[3]{\frac{[illegible]}{[illegible]}}$	$1.0 \times \sqrt[3]{48.31}$	45.70"	43.31"	
TRIPLE SCREW BATTLESHIP BISMARK	CENTER LINE	BOSS	$.80 \times \sqrt[3]{\frac{[illegible]}{[illegible]}}$	$.80 \times \sqrt[3]{20.418}$	25.34"	45"	
[illegible] TIRPITZ [illegible]	TWO OUTBD	STRUT	$1.0 \times \sqrt[3]{\frac{[illegible]}{[illegible]}}$	$1.0 \times \sqrt[3]{21.3152}$	33.20"	49"	
JAPANESE BATTLESHIPS YAMATO & MUSASHI	4	2 SKEG 2 STRUT	$1.0 \times \sqrt[3]{\frac{[illegible]}{[illegible]}}$	$1.0 \times \sqrt[3]{23.5656}$	34.4"		
[illegible]		DITTO ORIGINAL DESIGN	$1.0 \times \sqrt[3]{\frac{[illegible]}{[illegible]}}$	$1.0 \times \sqrt[3]{30.40662}$	37.45"		
* NOTE: The Narvik class tip clearance deficiency resulted in a "very rough" ship. The vertical vibration amplitude, measured at the fan-tail, amounted to 80 mils (double amplitude) at blade frequency and with the propellers turning only 270 RPM (38% full power).				$\sqrt[3]{}$			
				$\sqrt[3]{}$			

←GERMAN WARSHIPS→

TABLE II

ula which is posted in Table II. This test will limit the positioning of marine propellers to men capable of accurately multiplying and dividing numbers having magnitudes comparable to our national debt, and possessing the ability to extract a cube root of discrete magnitude.

Table II posts the principal hull dimensions, the power plant data, the full power speeds, and the propeller blade tip clearance for a number of merchant and foreign Navy ships and will afford a ready reference.

The tip clearance formula specifies not only the minimum propeller blade tip clearance for acceptable vertical and transverse vibrations but also determines the optimum position for the propulsion shaft alignment. The tip clearance dimension computed from the formula may be reduced about 5 per cent for propellers having the new NACABS hydrofoil sections. The NACABS design embodies a logarithmic mean camber line which results in a hollow pressure face with hooked leading edges at the outer radii of the propeller. The hooked leading edge causes a reduction in the hydrodynamic angle of attack and promotes shockless loading. Lowering the height of the mean camber line from the .95R to the tip accomplishes a reduction in the effective pitch which in turn causes an attenuation of the suction pressure field around the tips of the propeller blades. Hence the exciting force which causes objectionable vertical and transverse hull vibrations through resonant magnification is reduced about 5 per cent when compared with the conventional propeller design.

The factor of service, Z, posted in Table II embodies a consideration for reasonable comfort of passengers and crew in addition to fatigue failure of the hull structure and component parts. In a combatant ship it is essential to provide a steady gun platform, free from propeller excited vibrations, and also to permit the full usage of the delicate fire control instruments. The service factor is .75 for a destroyer rather than 1.0, because the propellers of a destroyer operate (and should operate) in the cavitation region at full power, and therefore the maximum and limiting suction pressure peak around the hydrofoil is attained prior to the full power output of the propulsion system. The working service factors may be computed by the proposed formula for existing ships possessing acceptable performance, and then this

service factor may be confidently applied for the positioning of the propellers during the design of similar ships, even though the hull size and installed S.H.P. is increased many fold.

The propeller tip clearance requirement parallels many other of Mother Nature's balanced regulations in that "too much" results in concealed prodigality whereas "too little" blazes a trail of distress.

From the perspective of the marine engineer it appears to be in order to suggest adding an appendage to General Forrest's theory. The Naval version of the formula for battle victory then becomes: "To get there fustest with the mostest, fit the fittinest propellers and keep the bearings loaded."

METALLURGICAL TRACERS AND NAVAL RESEARCH.

By Julius J. Harwood*.

One of the most important peacetime consequences of the nuclear energy developments of the past decade is the increasing availability of artificial radioactive isotopes. Not only have the radioisotopes which were previously known become more plentiful, but also, as a result of cyclotron and atomic pile reactions, many are now available which were hitherto merely "paper reactions." The major applications of radioactive tracers have been in the fields of medicine and biology, but there are numerous problems in metallurgy as well, which lend themselves to attack by means of radioactive tracer techniques. Many of the common metals have one or more radioactive isotopes with a satisfactory half-life and an adequate radiation intensity to permit their use as "tagged" atoms for metallurgical tracer purposes. The increasing availability of these isotopes has stimulated research along lines which hitherto resisted conventional methods of approach or necessitated laborious and painstaking efforts, often with uncertain results.

The problems related to the use of metals aboard the ships and aircraft of the Navy have been intensified by the accelerated advances of a modern fleet equipped with the latest technological and engineering developments. The "design" of super high-strength and tough alloys, the problems of corrosion and protection, the development of metals capable of withstanding extremely high temperatures, the understanding of the behavior of materials under service conditions—all of these are derived from and dependent upon advances in fundamental metallurgical knowledge. A constituent part of the basic research program of the Office of Naval Research is devoted to furthering and increasing the store of such knowledge in metallurgy. It seems natural that such a program will entail the use of radioactive isotopes in applying the most modern scientific techniques toward the accomplishment of its goal.

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Before describing the Navy's research programs and problems involved in the application of tracers to metallurgical research, a brief discussion of isotopes and their occurrence, properties, stability and means of detection may be worth while. An isotope has been defined as one of the several nuclides (atomic species) having the same number of protons in their nuclei and hence, belonging to the same element and occupying the same place in the periodic system. To phrase it somewhat more simply, isotopes of a chemical element are atoms having the same atomic number and chemical properties, but different number of nuclear neutrons and therefore different atomic weights (mass number). It should be noted that the generally listed atomic weight for a particular element is, in reality, an average weight of a mixture of stable isotopes.

Both stable and naturally occurring radioactive nuclei are known to exist for many of the elements, the former being by far the more abundant. The distribution of stable isotopes may be of interest. Elements of even atomic number usually have many more stable isotopes than those of odd atomic number, which in most cases have only one. The most abundant isotopes of even atomic numbered elements are usually of even mass number, while the abundant isotopes of odd atomic number are usually of odd mass number.

Naturally occurring radioactive isotopes are not stable but decompose on standing into other nuclei with the emission of energy. The greater stability of even atomic numbered elements holds true for the active isotopes also, since radioactive elements of even atomic number have longer lives than those of odd atomic number. The radiation emitted as a result of radioactive disintegration consists of three types; alpha (α) particle emission, beta (β) particle emission, and gamma (γ) rays. Alpha particles have a mass equal to that of a helium nucleus and a velocity approximately one-tenth that of light. They are positively charged with great ionizing power but relatively little penetrating power, and are only slightly deflected in an electrostatic or magnetic field. The beta rays, which consist of negatively charged particles, electrons, have speeds ranging from two-fifths to nine-tenths that of light. Their penetrating power is proportional to

their speed. They have less ionizing power than the alpha particles, but are more easily subjected to magnetic or electrostatic deflection. Gamma rays are identical with X-rays but have shorter wave-lengths, ranging from 1×10^{-8} to 1×10^{-10} cm. They obviously cannot be deflected by a charged field. The lighter weight radioactive elements usually emit electrons (beta rays). The heavier elements are beta and gamma ray emitters and the heaviest elements emit alpha particles as well beta and gamma rays.

Artificial radioactive isotopes are experimentally produced by the absorption of high energy particles or radiations with subsequent transformation. The transformation products may or may not be stable, and, if radioactive, disintegrate with the characteristic emissions described above. The source of the bombarding particles may be neutrons from uraniumplutonium piles; alpha particles, beta particles, protons or deuterons from accelerating devices such as the cyclotron or betatron; or the particles and rays from other natural or artificial radioactive sources. Typical reactions involving the transmutation of stable elements by bombardment are as follows:

$$B^{10} + \alpha^4 \rightarrow C^{13} + H^1 + \gamma^*$$

Stable boron + alpha particle → carbon + hydrogen + gamma rays
bombardment isotope isotope

$$C^{12} + P^1 \rightarrow N^{13} \rightarrow C^{13} + e^1$$

Stable carbon + proton → unstable nitrogen → carbon + positron
bombardment isotope isotope (positively charged electron)

The period of time necessary for the decomposition of a radioactive isotope is usually measured in terms of its "half-life," which is merely the time necessary for half of a given quantity of material to decompose. The radiation emitted as a result of the radioactive disintegration provides the means of detection and quantitative measurement of the radioisotope. A simple qualitative test for radioactivity is the action of these rays on photographic emulsions. However, long times of exposure are necessary when the activity of the sample or intensity of the radiation is low. Applications of this technique are more fully described

(* superscript number is mass number of isotope or particle).

later. The luminescent effect of these radiations upon such materials as zinc sulfide is also employed. For accurate and sensitive quantitative determinations the ionizing effects of these nuclear radiations upon gases are utilized. The most flexible and widely used instrument of this type is the well-known Geiger-Müeller counter. This is principally an ionization chamber containing an easily ionizable gas, e.g. argon, at low pressure, with two plates (a cathode and an anode) charged with a high electrostatic voltage. The radiation ionizes the gas, resulting in a flow of current which is a measure of the number of ions, which in turn furnishes a measure of the radioactivity. The voltage pulse produced is picked up and amplified by a vacuum tube circuit and the energy used to actuate a mechanical counter. A quenching gas or circuit is employed to quench each discharge very quickly so that the next signal may be received. It is possible to set a minimum voltage pulse below which the circuit will not count and thus count only alpha or alpha and beta type of radiation. In a Geiger counter, alpha radiation will always be recorded. The Geiger counter may be used to determine the number of disintegrations per unit time from an active sample, the type or types of radiation and the distribution of the particles in space. It is a valuable research tool, particularly when a mixture of radioelements is to be handled.

The well-known Wilson cloud chamber renders the path of the radiated particles visible by shooting them in a chamber containing supersaturated water vapor. The resultant ionization of the gas causes condensation of water droplets and the path of the particle can be easily seen and photographed. This type of apparatus has little application in tracer studies. It is mainly useful for research where it is necessary to follow the paths of single particles rather than measure the total amount of radiation from the source.

Stable isotopes can be detected and analyzed by the mass spectrograph, which separates atoms of different mass so that they can be recorded with a photographic plate or an ionization chamber. Stable isotopes have been used with a high degree of success in certain fields of endeavor, but their use in metallurgy has been severly restricted by the limited supply of concentrated mixtures available and the difficult means of detection with the

use of the mass spectrometer. The major advantage of using radioisotopes as tracers lies in the sensitivity of analysis and high degree of accuracy achieved. The limit of detection in a Geiger-Müeller counter is reached when the sample activity reaches the background count level which may be due to contamination by cosmic rays or the natural radioactivity of the specimen under examination. It has been reported that with the radioisotope C^{14} the activity can be followed through a dilution of 10^{10} fold. However, where the half-life of a radioisotope is so short as to make detection impossible, then it is necessary to resort to the use of stable isotopes.

As previously mentioned, isotopes of an element have identical chemical properties, and thus when small amounts of an isotope, be it stable or radioactive, is added as a tracer, the mixture behaves homogeneously in subsequent chemical or metallurgical reactions. This is the principle upon which the use of tracers is based. The important characteristics of radioactive elements which must be considered for their application to metallurgy are their half-life and energy of radiation. Usually the stronger the radiation, the shorter the half-life and vice versa. The half-life must be long enough for the metallurgical reaction to go to completion before disintegration is complete and at the same time the energy of radiation or the intensity must be sufficient for satisfactory detection. The half-lives of the tracers are also important in storing, stock-piling and personnel hazard considerations.

Foremost among the problems which indicated promise of fruitful endeavor by the application of radioisotopes was the study of diffusion processes, both volume and surface. A knowledge and study of diffusion coefficients is essential for a basic understanding of the kinetics and mechanisms of reactions in the solid state including such metallurgical phenomena as grain growth, surface and internal oxidation, age-hardening, phase transformations, etc. Thus as an essential part of its fundamental metallurgical program, the Office of Naval Research is cooperating with the Metals Research Laboratory of the Carnegie Institute of Technology and the Stevens Institute of Technology in the study of diffusion in the first major application of radioisotopes in metallurgy by university laboratories.

A large portion of the work at Carnegie Tech is devoted to

the study of the physical chemistry of steel-making. The needs for new high-strength and heat-resistant materials and at the same time the use of lower grade ores necessitate a thorough understanding of the reactions involved in steel-making in order to assure maximum production of steel of the requisite quality. The art of steel-making is still ahead of the science, and control methods have developed slowly, usually by empirical procedures. The variations in properties of armor plate and ship plate from heat to heat and from producer to producer lend sufficient testimony to this fact. The stringent demands of a modern Navy with more powerful weapons, increased fire power, new methods of power generation, heat and corrosion resistant materials, require a full and complete understanding of the fundamental chemistry and metallurgy involved in steel-making reactions, so that precise control on a scientific basis can be established.

Much of the prior work done to date on the problem of steel-making reactions had dealt with equilibrium relationships. This has been valuable and necessary but actual steel-making practice involves dynamic conditions in which reactions seldom attain the equilibrium state. In order to understand and control these reactions it is necessary to know the mechanism by which they occur and the rates at which they strive for equilibrium. Thus a clarification of the intrinsic differences among steel-making processes e.g. as between acid open-hearth and basic open hearth or between basic open-hearth and basic electric, may be developed. The program on molten metals at Carnegie is centered around the study of the mechanism and kinetics of reactions across slag-metal interfaces and with the measurement of activity of constituents in liquid metals and slags. The aim of the Carnegie researchers is to determine the rates and mechanisms whereby elements which are soluble in both slag and metal cross the interface between the two phases. Two elements, sulphur and oxygen, have been singled out for attention because of their primary importance in steel-making reactions.

The desulfurization of iron and steel in the various processes of manufacture from the blast furnace to the open hearth or electric furnace is becoming more and more important since our low sulphur grade ores are becoming exhausted. Aside from the use of sulfur in free-machining steels, it is desirable to maintain

the sulfur content of steels at a minimum. Excessive amounts of sulfur impart brittleness and red shortness and cause extreme difficulties in forging operations.

In the present study, radioactive iron and sulfur are being used to study the mechanism of transfer across a slag-metal interface and to determine the rate constants for the transfer of sulfur from slag to metal and vice versa. It is important to know whether migration occurs as a sulfur atom, a sulfide ion, a metal-sulfide compound soluble in both slag and metal or by a molecular interchange. Tagged atoms provide a direct way for determining the method of transfer, since their presence and path can be followed by the radioactive emanations.

Closely related to the problem of migration of sulfur across the slag-metal interface is the effect of such metals as calcium upon the mechanism of sulfur partition. It is well known that lime in the slag is an important factor in desulfurization and also in the oxygen distribution. It would therefore be enlightening to learn whether calcium is soluble to even a slight extent in the liquid metal phase, for it may provide a clue to intrinsic differences between steels made by acid or basic processes. The application of radioactive tracers to this problem is an excellent example of the utility of radioisotopes to metallurgical research. Conventional methods of analysis have not been sensitive enough to detect any solubility of calcium in steel but the use of tagged calcium atoms may provide a sensitive means of detection of low concentrations of calcium. But even if the activity is too low to be measured, it will still enable some estimate to be made of the maximum solubility of calcium in iron. Radioactive calcium is also being used to study the effects of slag composition and oxidizing conditions upon the calcium deoxidation constant in the metal phase, since any solubility of calcium in steel will vary with the degree of oxidation and with the slag basicity. The study of the mechanism of sulfur partition for both oxidizing (open hearth type) and reducing (blast-furnace type) slags may enable the establishment of a theory for sulfur partition applicable to any slag system and will aid in the interpretation of other slag-metal component reactions.

The problem of oxygen distribution, which may be of even greater importance, will follow the completion of the sulfur

studies. The absence of a suitable known radioactive isotope of oxygen makes this study somewhat more complicated. The oxygen transfer will have to be followed by the vacuum fusion analysis of quenched samples, but the methods of attack will be similar to the study of sulfur partition. The use of oxides containing radioactive iron and the determination of the transfer of active iron from slag into the melt will throw some light on the method of transfer of oxygen across the interface—whether it migrates as an iron oxide molecule or as oxygen atoms or ions.

Another phase of this research program is the study of the molecular constitution of slags. A knowledge of the dissociation of slags is important to the understanding of reaction rates and mechanisms. Previous work at Carnegie Tech has revealed that molten slags are at least partially ionized and at present electromotive force measurements, ionic mobility studies and conductivity studies are being used to elucidate the constitution and properties of slags. Also being studied are the effects of impressed electrical fields across the slag-metal interface upon the distribution of sulfur and oxygen between the slag and the metal and the mobility of the respective ions in the slag phase itself. The use of radioactive tracers in these studies will simplify the experimental problems involved. A simple application might be the immersion of two electrodes in a molten slag and following the migration of a particular tracer ion from one electrode to the other by simply bringing a Geiger Müeller counter close to first one and then the other electrode and measuring the differences in radiation intensity.

One of the most important and difficult problems of the present day is the task of predicting behavior of metals in service from laboratory and small scale tests. With Naval applications in particular, the problem is exceedingly complex because of the variety and degree of service conditions to which metals are subjected—and yet the operating limits are continually being pushed higher and higher. Naval engineers familiar with the problems of ship plate will appreciate the complexities of the problem. A vast amount of research has been concentrated, both during the war and at the present time on the development of a suitable small scale specimen test for predicting and interpreting ship plate behavior, and still the search goes on.

The significant mechanical properties of materials, namely tensile strength, ductility and energy absorption are not as simple as is generally supposed but must be considered in terms such factors as stress system, prior strain history, strain rate, stress and strain gradients, temperature, testing environment and the anisotropic nature of the internal structure. A specimen behaves in an anisotropic manner when its physical and mechanical properties are a function of direction; i.e. it has different properties in different directions. It is well known that the properties obtained from a simple uniaxial tensile test cannot be applied to a structural member where biaxial or triaxial stress conditions may exist. Nor can a simple notch-impact test such as the Charpy or Izrod test be used to predict the "toughness" and impact properties of a member when subjected to service conditions. In a simple analysis, the behavior of an infinitesimal region of a member under a combination of forces can be calculated and then translated to the member as a whole by the application of physical laws and geometric principles. But such exact solutions can only be obtained in those cases where the geometry of the member is quite simple and when the strain is within the elastic region. When the deformations are large and plastic flow occurs a factor of anisotrophy or directionality is immediately introduced into the problem as a result of the strain. The departure of the system from perfect isotropy is particularly important when combined stress systems are involved. Calculations and experimentation have shown that triaxial stresses have a great effect on the resistance to deformation, and yet the problems involved in the field of triaxial stresses are practically untouched. The entire problem of flow and fracture of metals is a field of research which at the present time is receiving a major concentration of scientific effort and talent, not only from the macroscopic point of view of the stress and strain relationships, but from a microscopic approach as well.

With further reference to the ship plate problem, it was early observed during the war that plates for merchant vessels which were ostensibly of the same composition and had received the same processing treatment, did not behave alike. The metallurgical factors responsible for the variability of ship plate properties are still a subject of research and may be related to the entire

history of the material from blast-furnace to the final heat-treatment. Obviously then, the behavior of metals under stress is intimately connected with the metallurgical changes which occur during the deformation of the metal. Many phenomena occur on a microscopic scale, such as phase transformations and precipitation reactions, which have an important bearing upon the distribution of stress and strain during plastic deformation. The magnitude and kinetics of the metallurgical reactions which may occur will be a function of the imposed stress system and in turn may affect the strain reaction of the system to such stress.

The strain at which a metal flows plastically will depend upon the rate at which the strain is applied. It is possible to apply a stress exceeding the yield strength fast enough so that little or no plastic deformation occurs. Thus the introduction of a high strain rate and large strain gradients localizes the deformation and prevents a redistribution of the stress such as occurs with slow strain rates. This redistribution is dependent upon the relaxation time and relaxation energy of the atoms. A knowledge of the coefficients of diffusion of the atoms, as a function of the strain, would aid in providing a better insight into this phenomena.

As a start toward this goal, the Stevens Institute of Technology is investigating the problem of "Diffusion in Stressed Metals" under Office of Naval Research contract. Starting with simple metals such as copper and zinc, determination is underway of the rates of self-diffusion of copper atoms through copper and zinc atoms through zinc, when these metals are placed under simple tension and compression. Measurements of the relationship between diffusion rate and strain within and beyond the elastic limit will be made at various temperatures. Copper and zinc were chosen because excellent experimental and theoretical data are available for these substances in a strain-free condition. Initial measurements on single metal crystals are being made first to provide information for interpretation of measurements on polycrystalline samples of the same substances. In this program also, since the study of self-diffusion of atoms is so difficult to follow by means of classical experimental techniques, "tagged" atoms of radioactive zinc and copper are being employed. Part of each crystal will be plated with a radioactive isotope and then heated to the desired temperature to permit diffusion to occur. Section-

ing the crystal and measuring the radioactivity of each section will permit determination of the rates of self-diffusion.

It is anticipated that this research program will throw much light on the role of diffusion in the recrystallization of binary alloy mixtures under action of stress and temperature variation and also yield information on internal changes in metals occuring under conditions of vibration.

It may be stated that the study of the kinetics of reactions in the solid state constitutes the basic science of the heat-treatment of metals and alloys, e.g. the hardening of steels on quenching, the tempering of steels, the age-hardening of duralumin and other non-ferrous alloys, because practically all of the metallurgical reactions in the solid state are dependent upon diffusion processes (the formation of martensite during the hardening of steels being a notable exception), it is obvious that a knowledge of the kinetics of diffusion is fundamental to an understanding of many metallurgical reactions. Such phenomena as hardenability, temper brittleness, surface oxidation and corrosion, recrystallization and grain growth and others too numerous to mention bear particular relevance to Naval applications of metallic materials and have concerned Navy metallurgists for many years. The development of high hardenability steels for armor plate ranging in size from thin sections to eighteen inch thickness, and for many other naval applications, depends upon a knowledge of the mechanism of hardenability and the functions of alloying elements in steels. The oxidation of metals at high temperatures, which has become so important for the development of gas turbines, pulse jets, rockets and the other new power plants is not as simple a process as one might imagine, but also is dependent upon the rates of diffusion of metal atoms through the oxide layers. It is only recently, as the science of physical metallurgy utilizing the fundamental laws of physical chemistry, thermodynamics and physics has developed and as our knowledge and understanding of these complex phenomena have deepened, that many of these problems appear to be on the verge of solution.

Thus, as another part of the program at the Carnegie Institute of Technology under office of Naval Research sponsorship, a comprehensive program devoted to the study of the "Kinetics of Reactions in the Solid State" is underway. This program is

concerned mainly with the rates of diffusion in solid metals and with concepts of the interatomic forces and activities which may be inferred from diffusion data. These studies are particularly susceptible to attack by use of radioactive tracers and the program is making extensive use of this technique. The various phases of this program under attack are as follows:

(*a*) *Kinetics and mechanism of the Austenite reaction*: At high temperatures, all steels exist with an austenitic microstructure. Austenite, which is a solid solution of carbon in gamma (face centered cubic) iron, when quenched rapidly from these high temperatures transforms into martensite, a solid solution of carbon in alpha (body centered cubic) iron. The acicular, needle-like plates of martensite which are formed are responsible for the high hardnesses and strengths or quenched steels. When cooled slowly, however, austenite decomposes into pearlite, which is a plate-like structure composed of alternate layers of ferrite (alpha iron) and cementite. Cementite is an intermetallic compound of iron and carbon and in itself is very hard and brittle. Pearlitic structures, however, are generally associated with soft or medium hard steels. The process of austenite decomposition is a most important one insofar as steels are concerned. It is intimately related to the self-diffusion of iron in both the gamma and alpha ranges and to the diffusion of carbon in gamma and alpha iron. The rates and coefficients of diffusion for these processes are being determined by the Carnegie scientists in the theoretical study of the mechanism of austenite decomposition. Special attention is being directed towards the effect of carbon concentration on the value of the diffusion coefficient of carbon in gamma iron. The self diffusion rates of iron have already been determined [1] and it has been found that the diffusion rate of iron in alpha iron is one hundred times greater than in gamma iron. The activation energy for the self diffusion of iron in alpha iron is also greater. Radioactive iron was used as a tracer in these studies.

In the determination of the coefficients of diffusion of carbon in iron, for example, a couple consisting of two steels of different carbon content is welded together. Following a diffusion heat-treatment in an inert atmosphere, transverse layers are machined

(1) R. F. Nehl & C. E. Birchenall—"Note on Self-Diffusion of Iron"—Mining & Metallurgy (AIME) November, 1947.

from the specimens and analyzed for carbon. From the resultant data, carbon penetration curves can be obtained and the diffusion constant can be derived by mathematical analysis.

(*b*) *Theory of Binary Diffusion*: Theoretical studies of published data on alpha brass resulted in a thermodynamic treatment which indicates that if the activities of the diffusing atoms are considered, rather than the concentration a coefficient may be calculated which is a constant independent of composition at constant temperature. [2] For substitutional binary solid solutions, three diffusion coefficients are involved, one for the self-diffusion of each atomic species and one for the "inter-diffusion" of the two diffusing atoms. This is a new concept of diffusion, and diffusion experiments in the gold-silver system are underway to substantiate the theory. Silver and gold were chosen since they form a completely miscible solid solution system. In addition, both metals can be obtained with an extremely high degree of purity and are not susceptible to oxidation.

(*c*) *Diffusion in Ternary Systems*: Most commercial steels do not consist of simple systems but are in reality complicated systems of four or more components. Even the simplest "carbon" steels may be regarded approximately as a ternary system of iron, carbon and manganese. The understanding of the behavior of these complex systems must be developed through a systematic approach and thus a generalization on ternary diffusion may then be applied to a variety of specific alloys.

A most important investigation underway in this phase is the study of the iron-carbon-boron system. This system is extremely interesting because of the important effects of boron on the hardenability of steels. Hardenability is the ability of a steel to harden in depth and is generally defined in terms of the diameter of a round which will just harden all the way through in an infinite quench. It is well known that the hardenability of steels is markedly affected by the alloy content but the explanation and mechanism of this effect are still unknown. As little as 0.003% boron increases the hardenability of a steel 20 to 30 percent, but it is not yet known whether boron is the only element which when added in such small quantities is able to exert such a large influence on hardenability. Thus the effects of boron on the diffu-

(2) C. E. Birchenall & R. F. Nehl—Thermodynamic Activities & Diffusion in Metallic Solid Solutions—Metals Technology 14 June 1947, T. P. 2168 (AIME).

sion of carbon, iron and other elements in steels may offer an insight into its potent but still mysterious role in the hardenability of steels.

(*d*) *Oxidation Studies*: Radioactive tracer techniques are extremely adaptable to the study of diffusion mechanisms in oxide films and thereby the mechanisms of the scaling of metals can be determined. Bardeen, Bratton and Shockley [8] studied the oxidation of copper by tracer techniques and demonstrated that the oxide grows by formation of vacancies in the copper lattice from the outer surface into the metal into which the copper ions diffuse. By the use of similar techniques, it is thus possible to determine whether it is iron or oxygen which diffuses through the oxide layer during the oxidation of iron. A thin layer of radioactive iron is electroplated on a polished surface of pure iron and subsequently oxidized under controlled conditions. The distribution of radioactive iron through the scale indicates the nature of the diffusion process. The initial results of the Carnegie experiments have already indicated that the generally accepted mechanism of the oxidation of iron is not sufficiently adequate. It is hoped that the information derived from these studies, together with the kinetic data now available in the literature will permit a complete reconsideration of the mechanism and kinetics of the scaling of iron.

Previously, it was indicated that one of the methods for detection of radioactive emanations is based upon the effect of the emitted rays upon photographic emulsions. Such contact film methods, known as autoradiography, are valuable for determination of distribution of impurities, microscopic layer concentrations, grain boundary reactions, carbon concentration gradients and other similar phenomena important to the study of solid reactions and metallic properties. A part of the Carnegie program is devoted to the development of such "nuclear microscopy" techniques for the study of these effects. First to be undertaken is the study of the distribution of carbon in the pearlite transformation. Iron surfaces, carbonized with radioactive carbon, have been treated to produce large-grained coarse pearlite, and fine grained photographic plates sensitive to soft beta radiations are being used.

(8) J. B. Bardeen, W. H. Bratton & W. Shockley—Physical Review *70*, 105, (1946).

It is interesting to note that focusing effects of electric and magnetic fields can be utilized to produce sharp, magnified images as in the electron microscope. L. Martin of the National Bureau of Standards with the cooperation of P. H. Abelson of the Carnegie Institute of Washington has developed a method using such a magnetic focusing arrangement in which the radiation given off by a radioisotope within a sample material is made to form an image of the emitting surface upon a photographic plate. This "Tracer micrographic" method has attained a resolution as fine as 30 microns.

The foregoing has been an attempt to demonstrate the applications of radioactive tracers to fundamental metallurgical research under the sponsorship of the Office of Naval Research. The major emphasis is on diffusion studies because of the paramount importance of such studies for the interpretation and understanding of many of the complex metallurgical phenomena which require clarification. But the above studies by no means represent the complete and ultimate application of radioisotopes in metallurgical research. The use of tagged atoms can lead to a better understanding of such commercial processes as nitriding, chromizing and siliconizing. The mysteries of the phenomenon of sintering in powder metallurgy may possibly be unravelled. Surface reactions, catalysis, adhesion and the oxidation resistance of stainless steels and other alloys can be studied. The mechanisms of metallic corrosion, which are of prime importance to the Navy, can be determiend. Friction, wear, ingot segregation, grain growth, recrystallization, phase transformations are all likely topics for research using radioisotopes.

These represent a few of the many problems which are of concern to Navy metallurgists and which can readily be attacked by tracer techniques. As these techniques become more developed, as personnel and handling problems become standardized, as general familiarity is attained, then one may expect large scale applications of radioactive isotopes throughout the nation's research laboratories, with a consequent acceleration of research progress.

DISCUSSION BY MR. F. NETTEL OF THE PAPER "GAS TURBINE PERFORMANCE CHARACTERISTICS, DESIGN CONSIDERATIONS, AND APPLICATIONS." BY J. T. RETTALIATA.

This paper appears very instructive to those readers who want fundamental information on what the gas turbine is like, what it can do today, and what it may do in the future.

However, certain statements invite clarification which is submitted in the following discussion.

1. The statement that "the basic gas turbine plant enjoys the added unique advantage of being independent of geographical location" does not appear to take into consideration that changes in both the temperature of the air at the compressor intake and the air density at this point, which are consequences of climate or altitude or both, have a profound influence on the performance of every gas turbine, including the "basic".

2. The data chosen for cycle E, especially the size of the heat exchanger alone with 5 sq. ft. per useful hp, are giving a somewhat too favorable picture, if we consider that modern h.p. steam plants often need not as much heat transfer surface per hp for the whole plant, including boiler, feed heaters, condenser, etc.

3. Comparison of cycles: The ideal Carnot cycle with 100 per cent efficiency of compression and expansion is being compared with cycles in which the turbine has an efficiency of 86 per cent and the compressor 84 per cent. The big difference shown results from the difference in component efficiencies, and not from any particular merit or demerit of the cycle as such. It should be noted that, if the same efficiencies are assumed for the rotating machines in the Carnot cycle, the efficiency comes out lower than that of cycle E, provided the latter is laid out for true optimum performance.

4. Refinements of cycles (inter-cooling and reheating in particular) do not only lead to better efficiencies, but what is of the greatness economic importance, also to smaller "air rates" (mass

flow of air per useful hp) i.e. to smaller machines per useful hp than required by the basic cycle.

5. "Optimum pressure ratio" is in fact a very elusive figure which has been found to change very widely for changes in the basic working data (component efficiencies, inter-cooling reheating, size of heat exchanger) so that the date given in Fig 8 should be used with caution. For instance for cycle E, a plant for 2000 hp and one for 10,000 hp will have widely different optimum pressure ratios due to the difference in component efficiencies; even more pronounced will be this difference in ratios if we choose a heat exchanger of 5 sq. ft/hp in one case and 2 sq. ft./hp in another.

6. Applications.—Rail: It seems that a somewhat too favorable picture is drawn for the gas turbine. If we admit the use of pulverized coal in a gas turbine, with some fly ash passing through the turbine, it is only fair to credit the Diesel engine with the possibility—and probability of similar hazards—of using lower grade fuel oils as fuel. Actually, extensive tests are under way on a British Diesel ship using furnace oil, with promising results.

More advanced types of Diesel engines, now under development, promise radical reduction in weight per hp, improved torque-speed characteristics, and even the possibility of starting under load. It can, therefore, be said that the gas turbine will have to be even better than pictured (except perhaps for the largest outputs), if it is to compete successfully with the Diesel locomotive.

The locomotive shown in Fig. 15, with hydro-mechanical transmission, suffers from one, not unimportant, disadvantage. The design of a hydro-mechanical transmission within the space limitations indicated ($10\frac{3}{4} \times 6 \times 5\frac{1}{2}$ ft.) presents a hard and risky design proposition (oil cooling, etc.) of which a really satisfactory solution is not known to the writer.

Aircraft: The fuel consumptions stated per lb static thrust, about 1 to 2 lb at sea level, look forbidding enough. It should, however, be remembered that the actual consumption is substantially lowered during flight at very high speeds and at very high altitudes, for which such gas turbines are principally—if not exclusively—suited.

NOTES.

INDEX TO NOTES.

FIRST MARINE GAS TURBINE IN SERVICE.
—The British Motor Ship, October, 1947.

ATOMIC-POWER ENGINEERING.
—Mechanical Engineering, September, 1947.

THE FUNDAMENTALS OF SHIP VIBRATION.
—The Shipbuilder and Marine Engine Builder, May, June, July, 1947.

THE LAW OF SIMILITUDE FOR FLOW PROBLEMS.
—The Sulzer Technical Review, No. 1, 1947.

FIRST MARINE GAS TURBINE IN SERVICE.

Reprinted from "The British Motor Ship" for October, 1947.

In order to obtain experience in the possibilities of the propulsion of ships by gas turbines, the British Admiralty selected a motor gunboat and arranged for the installation of a Metropolitan-Vickers unit on one of the three propeller shafts in the vessel, which is of a class used for the transportation of ball bearings from Sweden during hostilities. Before describing the new machinery, it is necessary to give an idea of the dimensions of the craft, bearing in mind that the new gas turbine is designed to develop about 2500 Shp., supplementing the power of the wing engines, which are Packard petrol motors, each capable of an output of 1250 Bhp. Thus, the total power installed in the boat amounts to 5000 Shp.

She has a length of 117 ft. overall and is 110 ft. on the waterline, the breadth being 19 ft., and the mean draught 4 ft. 3 ins. The displacement is 100 tons and the vessel was built by Camper and Nicholsons, Ltd., Gosport. In the installation of the new machinery, the builders co-operated with the Metropolitan-Vickers engineers, and the entire work was carried out in association with the Admiralty, the result being the first appearance afloat of a gas turbine for any purpose whatsoever. The engine-room plans are published on page 68.

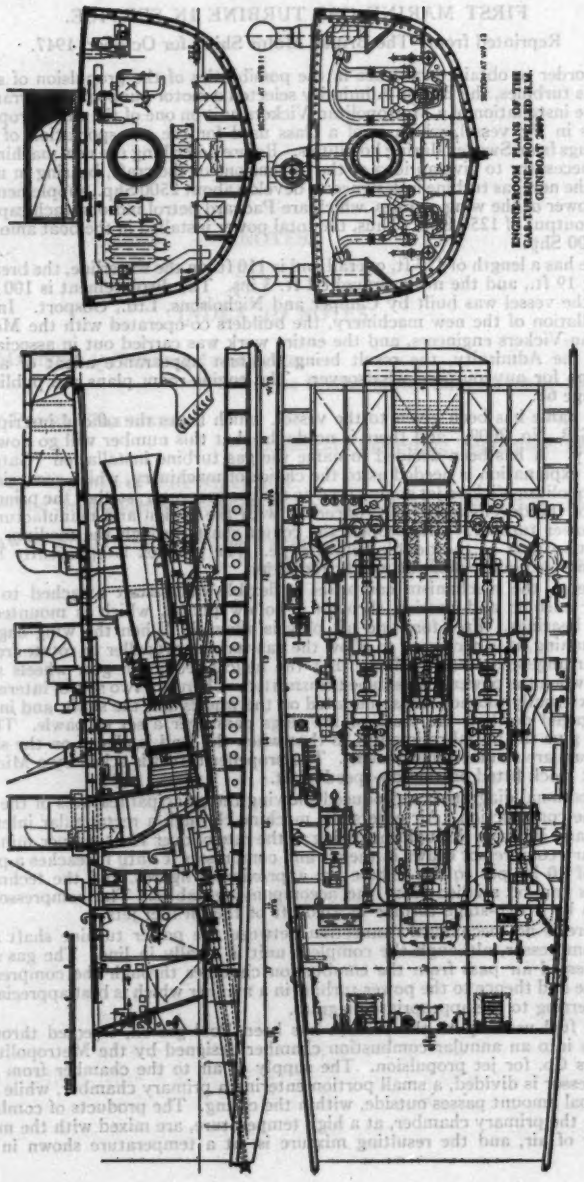
No name has been given to the vessel, which bears the official inscription "M.G.B. No. 2009," and there is no doubt that this number will go down in history. It has been decided to name the gas turbine installation "Gatric." Some explanation is needed as to the choice of machinery, which comprises a Metropolitan-Vickers jet engine used as the gas generator, so that the principal part of the development was concerned with the design and manufacture of the power turbine. This propels the center shaft through the medium of a reduction gear of the double-helical type. The gearbox is apparently built with no particular effort at weight reduction.

Briefly, the mechanism comprises a flexible quill shaft attached to the power turbine and passing through a hollow pinion, which is mounted in roller bearings. Free fore and aft play is allowed. When the wing engines are running, it is necessary to allow the gas turbine propeller to rotate freely, and arrangements are made so that the helical reduction gear-wheels slide apart when the power ceases to be transmitted. There are two sets of internally and externally toothed rings, mounted on the wheels and the shaft, and in the disengaged position the teeth of the rings pass over a set of pawls. These pawls give the initial resistance, so that when the load is placed on the shaft the gears are run into engagement. The propeller thrust is taken by a Michell thrust block fitted on the low-speed shaft.

A sectional diagram is reproduced showing the principal features of the gas turbine construction. Air enters the machine through a rectangular inlet on deck, and it passes through the casing at the rate of over 1¼ tons per minute. A rotary compressor draws in the air and compresses it until it reaches a pressure of 40 lb. per sq. in. These are approximate figures, and the technical details can be studied from the accompanying tables. The compressor is driven by a two-stage turbine located aft of the oil fuel jets.

There is no mechanical connection between the power turbine shaft and the compressor, although the complete unit is axially in line. The gas and compressed air pass from the combustion chamber through the compressor turbine and thence to the power turbine in a manner which is best appreciated by referring to the appropriate diagram.

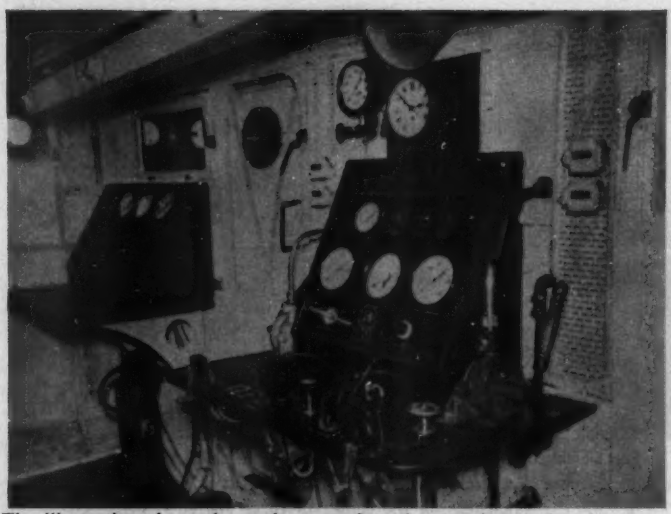
The fuel used up to the present has been Pool gas-oil, injected through 20 jets into an annular combustion chamber designed by the Metropolitan-Vickers Co. for jet propulsion. The supply of air to the chamber from the compressor is divided, a small portion entering a primary chamber, while the principal amount passes outside, within the casing. The products of combustion in the primary chamber, at a high temperature, are mixed with the main supply of air, and the resulting mixture is, at a temperature shown in an



ENGINE-ROOM PLANS OF THE GAS-TURBINE-POWERED H.M. GUNBOAT 2009.

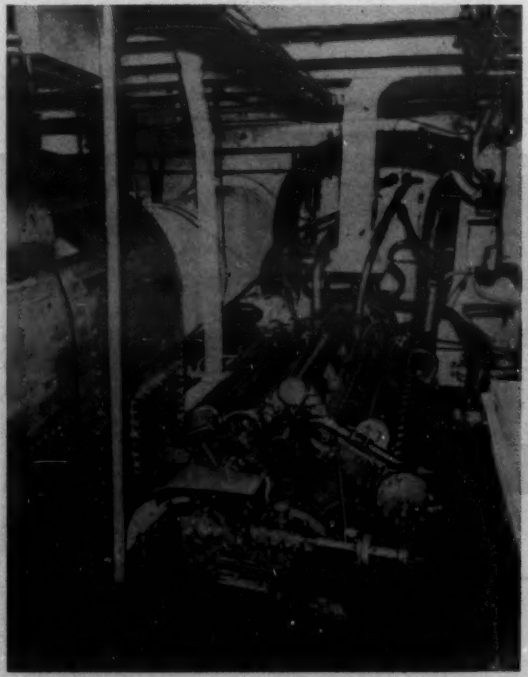
The first vessel to put to sea with a marine gas turbine, H. M. M. G. B. 2009, built by Camper and Nicholsons Ltd. and fitted with a Metropolitan-Vickers machine on the center of three shafts.

Door removed at side of casing, showing horizontal control rod for 2500 S.H.P. gas turbine.



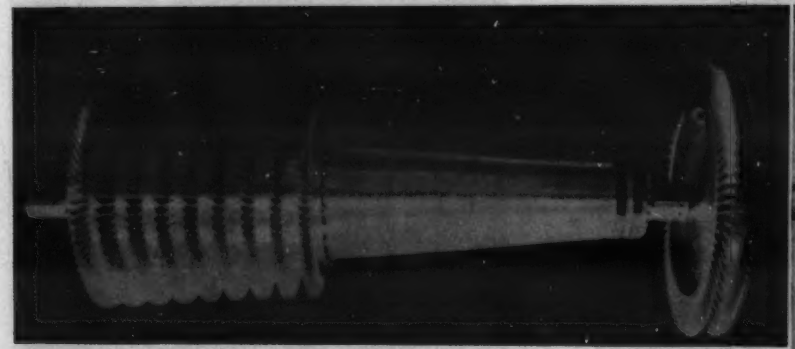
The illustration above shows the controls and gauge boards for the center shaft, propelled by the gas turbine.

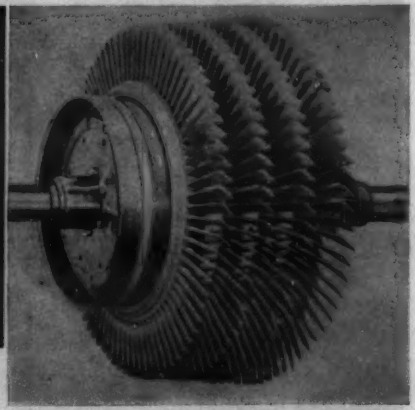
The main electrically driven ventilating fan on deck.



The intake end of the turbine is shown to the left in the view above, one of the wing engines being seen alongside.

End of the turbine, showing the exhaust branch which leads to the funnel.





Compressor and two-stage driving turbine unit (above). This has no mechanical connection with the main power turbine (right).

accompanying curve, sufficiently reduced for admission to the power turbine blading. A considerable amount of information can be obtained from the curves to which we refer, and these are actually of greater use than an extended description of the action.

The jet engine does not give the best possible performance from the point of view of maximum efficiency. The machine is, however, a proved and tested unit, and some sacrifice was deliberately made in order to obtain the necessary overall results with the complete propelling unit with a minimum amount of delay. The fuel consumption is relatively high. In a Naval vessel of the type in question, the full power of the center shaft operated by the gas turbine will be required only at more or less rare intervals, and the overall efficiency is not of great importance. It may be noted that the lowest consumption hitherto obtained is 1.06 lb./Shp. hr. on test, using Pool gas-oil with a calorific value of 19,600 Btu.'s/lb. gross, equivalent to 18,560 Btu.'s/lb. net. One of

General Conditions on Trials

	Passage Area	Approx. Velocity	Temperature	Gauge Pressure	Total Head
Atmospheric conditions	—	—	Approx. 70°F	0	146 lb./sq. in.
Entrance to bridge intake	24 sq. ft.	26 ft./sec.	70°F		
Entrance to down-duct	8 sq. ft.	78 ft./sec.	70°F		
Middle of down-duct	8 sq. ft.	78 ft./sec.	70°F		
Exit from down-duct	8 sq. ft.	78 ft./sec.	70°F		
Intake into Bakewood trunk	—	—	70°F	Max. suct. head, 10"	14.47 lb./sq. in.
Middle of Bakewood trunk	4 sq. ft.	156 ft./sec.	70° + 70°		
Entrance to axial compresser	—	—	Abt. 200°C	37 lb./sq. in.	45.7 lb./sq.in.
Exit from axial compresser	—	—	390°F		
Entrance to compressor turbine	—	—	800°C 1472°F	33 lb./sq. in.	41.9 lb./sq. in.
Exhaust from compresser turbine	—	—	(690-1,275 730°C 1346°F	8 lb./sq. in.	23.34 lb./sq. in.
Entrance to power turbine	—	—			
Exhaust from power turbine	—	—	580°C 1076°F	0.2 lbs./sq.in.	15.525 lb./sq. in.
Top of inner funnel	—	350 ft./sec.	500°C 932°F	0	—

Air flow at full R.P.M.
617.5 cub. ft./sec.
37,050 cub. ft./min.

At 13.5 cubic ft. per lb.
47.5 lb. weight/sec
2,850 lb. weight/min
1.27 tons/min.
76 tons/hr.

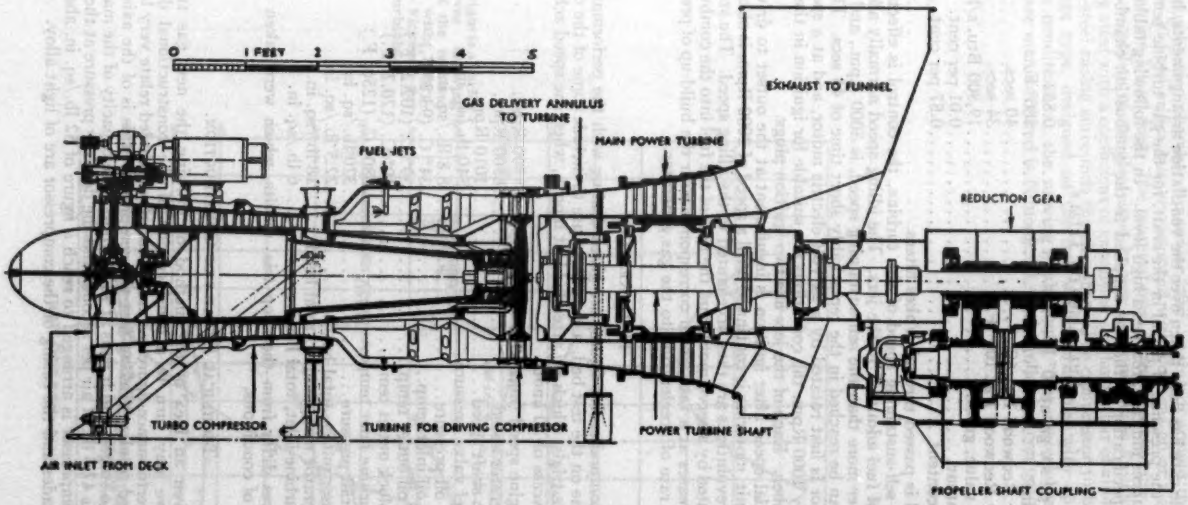
the most important points to observe is that the weight of the unit, including the gearing, does not greatly exceed 2¼ lb. per Shp., which is less than that of the wing engines running at 2400 Rpm. The speed of the gas turbine is 3600 Rpm., and the propeller shaft is driven at one-third of this figure.

Propeller Dimensions.

The wing propellers, driven by the reciprocating engines, have a diameter of 44 ins. and a pitch of 63 ins. They are of the three-bladed type with an area of 815 sq. ins., the speed for which they were designed being 800 Rpm. The center propeller, driven by the gas turbine, also has a diameter of 44 ins., the pitch being 46 ins. The speed for which this propeller was designed is 1087 Rpm. All the propellers are of manganese bronze, the port screw being left-handed and the starboard screw right-handed, while the center shaft is left-handed, machined to a diameter of 4 ins. from stainless steel. The wing shafts have a diameter of 3.5 ins. and are of Monel K metal.

Example of a Shop Trial

Gas Generator	Speed	6,850 R.P.M.
	Trial pressure	352 lb./sq. in.
	Outlet temp.	680°C
	Compressor inlet : Total head	14.473 lb./sq. in. abs.
	Vel. head	1.87 lb./sq. in.
	Compressor outlet : Total head	45.7 lb./sq. in. abs
	Pressure ratio	3.16
	Temp. rise	123°C
	Turbine inlet : Total head	41.9 lb./sq. in. abs.
	Combustion chamber, pressure drop	3.8 lb./sq. in.
Power Turbine	Shaft speed	1,082 R.P.M.
	B.H.P.	2,261
	Inlet : Total head	22.34 lb./sq. in. abs.
	Static	20.78 lb./sq. in. abs.
	Outlet : Total	15.525 lb./sq. in. abs.
	Static	15.000 lb./sq. in. abs.
	Pressure ratio	1.55
	Turbine temperature drop	97°C
	Outlet temperature	580°C
	B.H.P.	2,259
	Specific consumption	1.163 lb./a.h.p. hr.



Longitudinal section of turbo-compressor, combustion chamber, turbine driving compressor and independently mounted power turbine for driving the propeller shaft through reduction gear.

Considerable care has been taken over the ventilation arrangements, having due regard to the presence of petrol in the craft, for the gas turbine temperature, as the curves indicate, reaches a high level. In the following table is an abbreviated specification of the fuel supplied to the combustion chamber.

Fuel Used on Trials.

Pool gas-oil sp. gr. at 60° F.	0.832
Flash point	180° F.
Viscosity Redwood I at 60° F.	40 secs.
Viscosity Redwood I at 100° F.	34 secs.
Calorific value, gross	19,800 Btu.'s/lb.
Coke content	0.01 per cent.
Sulphur content	0.87 per cent.

All the fuel is passed through Vokes filters.

Above the self-sustaining speed of the turbine, the control is effected by the amount of fuel admitted to the jets. The idling speed normally allowed, which is rather more than the self-sustaining speed, is 3000 Rpm., and these revolutions can be reached in the remarkably short time of 45 secs. The gas generator rotor is first rotated by a 24-volt electric motor, and at a speed of approximately 1000 Rpm. the conditions are suitable for ignition in the combustion chamber. Each of the jets has two ignition plugs.

As the initial speed after ignition is insufficient at the outset to drive the compressor unit, motoring is continued until a speed of 2000 Rpm. is obtained. The turbine revolutions are then accelerated to the idling speed. The starting valve is operated by a servo motor and the flow of the fuel into the combustion chamber increases at a rate which corresponds with the build-up of pressure, giving a safe rate of acceleration to the gas generator.

Trial Results.

Although comprehensive figures are given dealing with the performance of the gas turbine on the test bed, it may be of use to include some of the details relating to the trials, which we attended last month, when the speed achieved was well in excess of 30 knots.

Wing engine speed (normal cruising rate)	1800 Rpm.
Gas generator speed	6500 Rpm.
Propeller shaft speed	1010 Rpm.
Fuel relief valve pressure	540 lb./sq. in.
Gearbox oil pressure	8.8 lb./sq. in.
Gearbox oil inlet temp.	34° C. (93.2° F.)
Gearbox oil outlet temp.	39° C. (102.2° F.)
Thrust block outlet temp.	49° C. (120.2° F.)
Power turbine inlet temp.	680° C. (1256° F.)
Fuel jet rail pressure	320 lb./sq. in.
Compressor outlet, total head	22.5 lb./sq. in.
Gas generator turbine inlet, total head	18 lb./sq. in.
Power turbine inlet, total head	6 lb./sq. in.

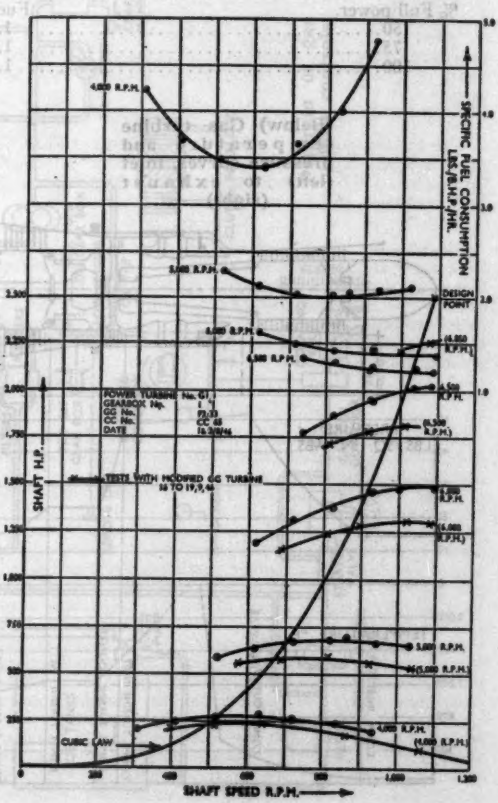
These figures differ from the shop trial results, which were taken under different sets of conditions.

Technical Details of Construction.

Having given an idea of the method of working the new gas turbine machinery, we may turn to a consideration of the constructional details. Such of the performance figures as are necessarily included relate very largely to the test bed results. Taking first the compressor, this is of the axial-flow type with nine stages, and it runs at 7300 Rpm. The capacity of the machine, as designed, is 47.5 lb. of air per second, and the delivery pressure at the inner end of the compressor is arranged to reach a figure of 52 lb./sq. in. absolute. The rotor, blades and the casing of the compressor are of light alloy.

The combustion chamber is annular, and, as stated, around the circumference are 20 fixed orifice burners, supplied with Pool gas-oil or a good quality of submarine Diesel fuel. The material used in the manufacture of the flame tube is a quality of stainless steel known as Immaculate 5. This is a heat-resisting steel with a coefficient of expansion similar to that of aluminum alloy. A mild-steel casing surrounds the chamber. The compressor turbine is of the two-stage type, designed on the free vortex principle, with approximately impulse conditions at the moving blade roots. The material of which the blades are manufactured is Nimonic 80, of the following composition:

Ni	71
Cr	22
Fe	3.0
Ti	2.5
Cu	0.1
Ac	0.4
C	0.05
Mn	0.35
Si	0.70



On the right is a series of curves showing some of the test-bed results with Metropolitan-Vickers gas turbines.

The power turbine is of the four-stage design, on similar lines to the compressor turbine. The drum and blades are of Molyvan and, the cylinder, which is of mild steel, is nickel coated internally. The rated output is 2600 Shp. at 3600 Rpm. The complete unit, including the gearbox, weighs 7075 lb. or 2.80 lb./Shp. At 2500 Shp. the fuel consumption is 1.06 lb. per Shp./hr.

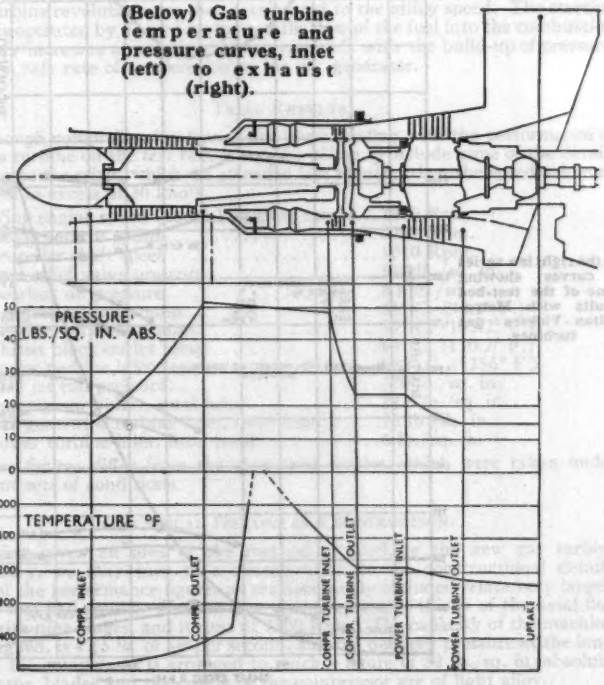
A table is included giving the fuel consumption and power at different speeds of the gas generator.

FUEL CONSUMPTION AND POWER ON TEST.

Gas generator speed.	Bhp.	Fuel consumption lb./Bhp./hr.
4000	270	3.48
5000	670	2.07
6000	1460	1.42
6500	2000	1.2
7220	2500	1.06 (Test only)

% Full power.	Fuel consumption.
50	1.7 lb./Shp./hr.
75	1.4 lb./Shp./hr.
100	1.2 lb./Shp./hr.

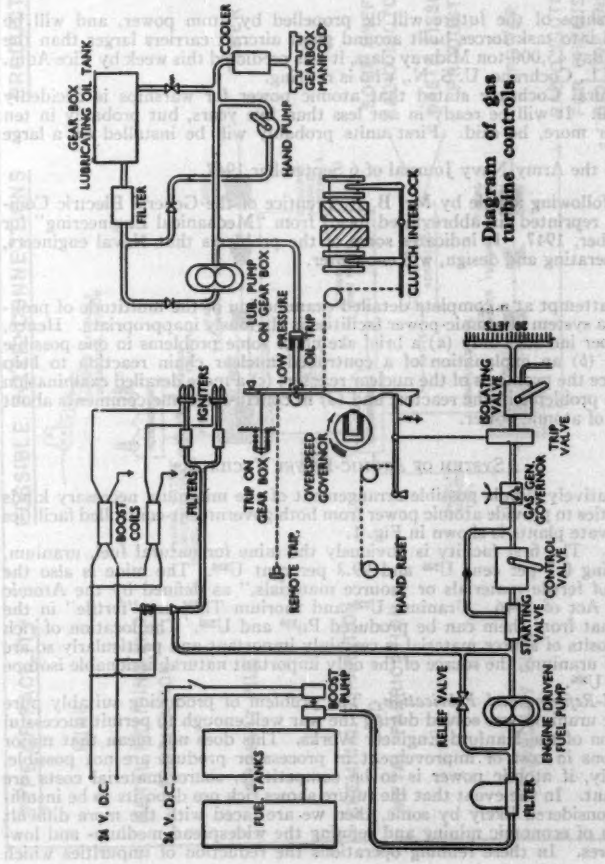
(Below) Gas turbine temperature and pressure curves, inlet (left) to exhaust (right).



The general conditions noted during the test bed trials are shown in the upper table.

The compressor weighs 1660 lb., and the turbine 2050 lb., making an engine weight of 3710 lb. This compares with a Packard engine weight of 2210 lb. The gas turbine gearbox weighs 2600 lb., which is more than that of the turbine, while each Packard engine gearbox weighs 700 lb. The supporting frame of the gas turbine has a weight of 230 lb., and the lagging, which is an essential feature of the installation, weighs 290 lb. The controls and auxiliaries for the gas turbine have a total weight of 90 lb. The lubricating oil tanks weigh 200 lb., and the weight of the oil in these tanks is 450 lb.

Based on an output of 1800 Shp., the first gas turbine unit has a weight of 6310 lb., with the gearing, equivalent to 3.5 lb. per Shp Later units, with an output of 2500 Shp., turn the scale at 6700 lb., or 2.68 lb. per Shp., and this figure may be compared with 2.93 lb. per Shp. for the wing engines. It may



be of interest to add that the expansion, at full throttle, of the gas turbine unit is in the neighborhood or ⅛ of an inch per linear ft.

In the lower table is a further set of figures relating to one of the test bed trials.

It should be added that the Admiralty proposes to extend this development by installing a higher-powered unit in an escort vessel. The use of petrol on board will no doubt be discontinued, in our opinion, and if this should be the case the need for elaborate ventilating arrangements will become of less importance than with the first experimental craft.

ATOMIC-POWER ENGINEERING.

Some Design Problems.

"Warships of the future will be propelled by atom power, and will be grouped into task forces built around giant aircraft carriers larger than the present-day 45,000-ton Midway class, it was predicted this week by Vice Adm. Edward L. Cochrane, U. S. N., who is retiring.

"Admiral Cochrane stated that atomic power for warships is decidedly practical. It will be ready in not less than five years, but probably in ten years or more, he said. First units probably will be installed on a large carrier."

From the Army-Navy Journal of 6 September 1947.

The following article by Mr. B. R. Prentice of the General Electric Company is reprinted (in abbreviated form) from "Mechanical Engineering" for September, 1947. It indicates some of the problems that Naval engineers, both operating and design, will encounter.

Any attempt at a complete detailed examination of the multitude of problems in a system of atomic-power facilities is obviously inappropriate. Hence, this paper includes only (*a*) a brief sketch of some problems in one possible system; (*b*) an explanation of a controlled nuclear chain reaction to help introduce the problems of the nuclear reactor; (*c*) a more detailed examination of some problems in the reactor; and (*d*) in conclusion some comments about growth of atomic power.

System of Atomic-Power Facilities.

A relatively simple possible arrangement of the minimum necessary kinds of facilities to provide atomic power from both government-controlled facilities and private plants is shown in Fig. 1.

Mine. The first facility is obviously the mine for natural fuel, uranium, containing 0.7 per cent U^{235} and 99.3 per cent U^{238}. The mine is also the source of fertile materials or "source materials," as defined by the Atomic Energy Act of 1946. Uranium U^{238} and thorium Th^{232} are "fertile" in the sense that from them can be produced Pu^{239} and U^{233}. The location of rich ore deposits of source material is certainly important and particularly so are those of uranium, the source of the only important natural fissionable isotope known, U^{235}.

Metal-Refining and Fabrication. The problem of producing suitably pure metallic uranium was solved during the war well enough to permit successful operation of the Hanford Engineer Works. This does not mean that major reductions in cost or improvement in process or product are not possible. Certainly, if atomic power is to be competitive, source-material costs are important. In the event that the future shows rich ore deposits to be insufficient, considered likely by some, then we are faced with the more difficult problem of economic mining and refining the widespread medium- and low-grade ores. In these refining operations the reduction of impurities which

absorb neutrons may be a difficult but important problem. Much remains to be learned of the metallurgy of uranium and thorium before the forming and fabrication of elements of the nuclear reactor can be accomplished with the ease and flexibility needed to permit optimum designs.

Primary Pile. In Fig. 1 source materials are shown being charged into the next basic facility, the "primary pile" or primary nuclear reactor. The concept of "primary" and "secondary" piles has been developed in negotiations with the United Nations. Primary piles contain source materials and produce new fissionable material; therefore it is expected that they may be under

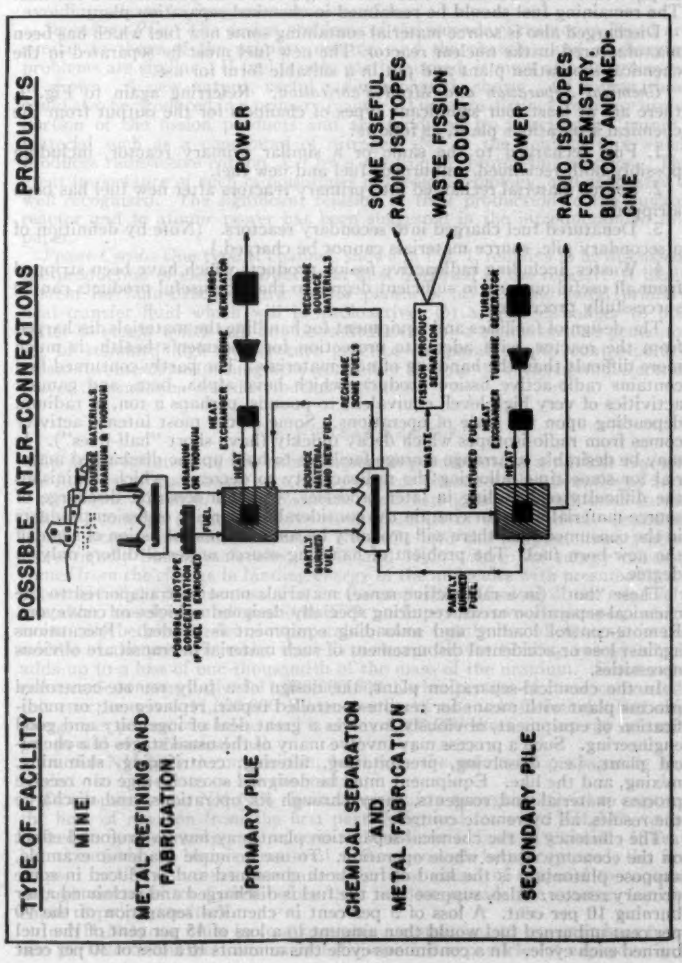


FIG. 1 ATOMIC-POWER FACILITIES

control of the government or of an international authority. Secondary piles do not, and it is expected that they may be privately operated, possibly with inspection.

It is possible that some primary nuclear reactors may use fuel enriched in U^{235}, so in Fig. 1 an isotope-concentration plant is shown which may or may not be required in the channel of fuel charged to the primary reactor.

Design problems of the reactors are discussed in later sections. In this system outline the interconnections should be noted. It is likely that fuel will be discharged before it is entirely burned. Obvious reasons are reduction in pile reactivity because of burn-out of fuel, economic balance, and the like. The remaining fuel should be reclaimed in chemical-separation plant.

Discharged also is source material containing some new fuel which has been manufactured in the nuclear reactor. The new fuel must be separated in the chemical-separation plant and put in a suitable form for use.

Chemical Separation and Metal Fabrication. Referring again to Fig. 1, there are at least four significant types of channels for the output from the chemical separations plant, as follows:

1. Fuel recharged to the same or a similar primary reactor, including possibly both reclaimed (unburned) fuel and new fuel.
2. Source material recharged into primary reactors after new fuel has been stripped.
3. Denatured fuel charged into secondary reactors. (Note by definition of a secondary pile, source materials cannot be charged.)
4. Wastes, including radioactive fission products which have been stripped from all useful outputs in sufficient degree so that the useful products can be successfully processed.

The design of facilities and equipment for handling the materials discharged from the reactor, with adequate protection for workmen's health, is much more difficult than the handling of new materials. The partly consumed fuel contains radio-active fission products which have alpha, beta, and gamma activities of very high level, equivalent to pounds, perhaps a ton, of radium, depending upon the scale of operations. Some of the most intense activity comes from radio-isotopes which decay quickly (have short "half-lives"). It may be desirable to arrange storage facilities to hold up the discharged material for some time, allowing the radioactivity to decrease, which diminishes the difficulty of handling in later processes. While a separate discharge of source material may not contain the considerable quantity of fission products in the consumed fuel, there will probably be some because of fission of some of the new-born fuel. The problem of handling source material differs only in degree.

These "hot" (in a radioactive sense) materials must be transported to the chemical-separation areas, requiring specially designed vehicles or conveyors. Remote-control loading and unloading equipment is needed. Precautions against loss or accidental disbursement of such material in transit are obvious necessities.

In the chemical-separation plant, the design of a fully remote-controlled process plant with means for remote-controlled repair, replacement, or modification of equipment, obviously involves a great deal of ingenuity and good engineering. Such a process may involve many of the usual stages of a chemical plant, i.e., dissolving, precipitating, filtering, centrifuging, skimming, mixing, and the like. Equipment must be designed so each stage can receive process material and reagents, carry through its operations, and discharge the results, all by remote control.

The efficiency of the chemical-separation plant may have a profound effect on the economy of the whole operation. To use a simple academic example, suppose plutonium is the kind of fuel both consumed and produced in some primary reactor. Also, suppose that the fuel is discharged and reclaimed after burning 10 per cent. A loss of 5 per cent in chemical separation of the 90 per cent unburned fuel would then amount to a loss of 45 per cent of the fuel burned each cycle. In a continuous cycle this amounts to a loss of 30 per cent

of all fuel charged. This example also highlights the related importance of designing the pile and fuel to permit a maximum percentage of fuel consumption before discharge.

The chemical-separation-process designer is faced with many auxiliary problems. Reagents may become contaminated with fission products either as a normal part of the process or by accident in the process. Fission-product wastes must be disposed of in a safe fashion. If they are to be stored, the volume of vehicle must be kept to a minimum. If they are to be shipped elsewhere, even more stringent requirements are placed upon concentration. Some of the fission products are gases. These must be handled and disposed, avoiding release in dangerous concentrations.

Secondary Pile. The design problems of the secondary reactor in Fig. 1 are similar to those of the primary reactor, and the charging and discharging problems are similar. If fuel is only partly burned, it must also be processed as shown. A new output, useful radioisotopes, is shown. Of course, these could also be produced in a primary reactor. These are distinct from any useful portion of the fission products and are produced by inserting a "parent" material such as a compound of nitrogen $_7N^{14}$ in the neutron flux which produces radioactive carbon $_6C^{14}$ by a neutron-in, proton-out reaction. The great importance of such radioisotopes in medicine, biology, and chemistry is well recognized. The significant relation of their production to the nuclear reactor and to atomic power has been suggested in the introduction to this paper.

Power Cycle. One type of channel, shown in Fig. 1, remains to be discussed, the production of electric power from heat liberated in the reactor. A likely system for land-based electric power plants is (*a*) a liquid-metal primary heat-transfer fluid which will be radioactive; (*b*) a heat exchanger to boil water to steam which will not be radioactive; (*c*) a steam-turbine generator set of standard high-temperature practice comparable with contemporary steam turbines transforming heat from other fuels; and (*d*) the normal electric-power transmission and distribution facilities.

The Nuclear Reactor.

In order to integrate the many engineering problems of a nuclear reactor, the function of each basic part in the nuclear chain reaction is outlined first.

Analogous Chemical Chain Reaction. Interesting as an introduction to the "chain" concept is the explosive decomposition of the relatively unstable, loosely bound, ammonium nitrate NH_4NO_3 into the simpler, more stable, more tightly bound molecules, N, O, and H_2O. Here the energy released comes from the change in binding energy of the molecules with presumably an extremely slight loss in mass. In the decomposition (fission) of the relatively unstable, loosely bound uranium U^{235} nucleus into two fragments, which eventually become nuclei of more stable, more tightly bound elements in the middle of the periodic table, there is a change in the binding energy which adds up to a loss of one-thousandth of the mass of the uranium. Since mass converted to energy gives 11,000,000,000 Kwhr. per lb. of mass converted, this reaction gives 11,000,000 Kwhr. per lb. of uranium fissioned. But this energy per pound of nuclear fuel is about 3,000,000 times the energy released per pound of a chemical fuel.

In the NH_4NO_3 decomposition, heat was the means of propagating the chain. In U^{235} neutrons are the means.

To propagate the NH_4NO_3 chain, one requires enough material to capture the heat of reaction from the first particle ignited for more NH_4NO_3 to be heated to ignition temperature, so it will continue to burn in a chain. If heat is lost outside the body of material without enough being captured, the fire stops. In U^{235} one must also have a "critical mass" of material to capture enough neutrons for fission before they are lost outside the reactor, for the chain to propagate itself.

In NH_4NO_3 one must have a favorable arrangement to transmit heat from one particle to another; one cannot disperse a few particles of explosive in a

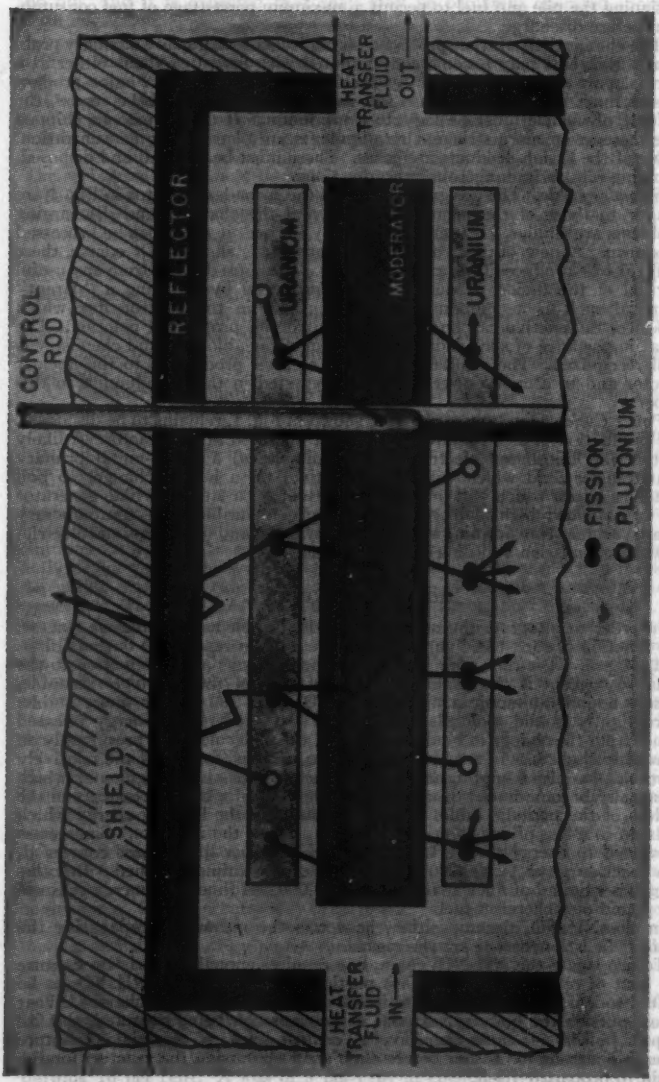


FIG. 2 NUCLEAR-ENERGY HEAT GENERATOR

pile of sand and make it explode. In U^{235} the amount of neutron-absorbing material in the reactor is very important and is generally detrimental to neutron economy. In nuclear reactors, however, neutron absorbers are sometimes used to slow or stop the reaction, much as if—by anology—one threw sand into a burning pile of NH_4NO_3 to quench it.

Controlled Nuclear Chain Reaction. In Fig. 2 is shown a schematic arrangement of the following:

1. Uranium fuel, containing U^{235} (fissionable material) and U^{238} (fertile material for production of Pu^{239}) in arbitrary proportions.
2. Moderator for slowing down neutrons.
3. Reflector for conserving neutrons.
4. Shield for protecting power-plant personnel.
5. Heat-transfer fluid for transporting the heat energy out of the nuclear reactor.
6. Control rod for varying the reactivity of the reactor and the power level.

In considering the chain reaction, it is convenient to analyze what happens in one generation of fissions. A generation may be described as the average length of time between when a neutron is liberated at fission and when it enters a nucleus of U^{235} to cause another fission. This length of time of average life of a neutron will be sensibly constant in any one design of nuclear reactor without regard to the power level. Therefore if the reactor is operating at constant power level, then the number of fissions in one generation is exactly equal to the number of fissions in the succeeding generations, as long as the power level remains constant.

In Fig. 2 are shown four fissions occurring during one generation, and, for simplicity, it is assumed that they all occur at the start of the generation and all in the upper fuel bar. Between 1 and 3 neutrons are liberated by each fission, so 10 neutrons are shown being liberated by the four fissions. The probability that a slow or thermal neutron will cause fission in U^{235} is very much greater than the probability that a fast neutron will cause fission. In other words, the U^{235} fission cross section is much larger at low than at high neutron energies. Therefore in many types of reactors, a moderator, whose function is to slow down neutrons, is placed around a fuel rod in such an amount that most of the neutrons will be slowed down to thermal energies before they have an opportunity to cause fission in a bar of fuel. Recalling the chemical analogy, this is simply one way of getting a "favorable arrangement."

At constant power level, exactly 4 neutrons from one generation of fissions must cause exactly four fissions in the next generation. Therefore in the diagram are shown 4 neutrons passing through the moderator where their direction is changed and energy reduced by collisions with nuclei of the moderator before they enter the lower fuel bar, where these 4 neutrons produce four fissions which in turn produce another 10 neutrons for the succeeding generation.

In this academic example, 6 neutrons are not required for fission. As many as possible of these extra neutrons should be used for productive purposes. One possible use is for the production of new fuel. Some U^{238} has been included in the U^{235} fuel bar, and the pile has been arbitrarily arranged with a proportion of U^{238} to U^{235} so that the probability that a slow neutron will be captured by U^{238} to form plutonium is equal to the probability that a slow neutron will be captured by U^{235} to cause fission. Therefore there are four new nuclei of plutonium formed in this generation by 4 neutrons slowed down in the moderator or reflector.

In any practical reactor there must be an allowance for loss of some neutrons. In Fig. 2 is shown one neutron escaping through the reflector into the shield where it is slowed down, then absorbed. It is lost in so far as any productive use is concerned.

If there is to be a means for intentional acceleration or deceleration of this chain reaction, positive means must be available to change the reproduction of the fissions in each generation from the 1-for-1 basis shown to a different reproduction ratio. Since the neutron flux is the medium of propagating the reaction, control of it can change the reproduction ratio of fissions. To accom-

plish this, a control rod which consists of a good neutron absorber such as cadmium or boron, is used. This control rod is shown extended into the pile just far enough to capture one of the 10 neutrons. If the control rod were in this same position but the power level happened to be greater or less than that shown, the control rod would still capture 10 per cent of the neutrons, thus maintaining the 1-for-1 reproduction even though the fission intensity was different. To accelerate the action simply withdraw the control rod and permit the neutron which it had previously absorbed to cause an additional fission. In this example it was arbitrarily assumed that the design and the ratio of U^{235} to U^{238} is such that the probabilities of a neutron-producing fission or making plutonium are equal. Therefore this change would, on the average, beget four and one-half fissions in the next generation for four in the first. The power level increases in a geometric progression. Since the period of a generation is constant, the acceleration is exponential with time and will continue indefinitely until something changes the ratio of reproduction. When the desired higher power level is reached, the control rod is returned to approximately its previous position or where it captures 10 per cent of the neutrons. Then the reactor is operating at a new constant but higher power level. A reverse procedure is used to reduce power level.

In addition to accomplishing the dynamic control of the reactor for purposes of applying it to power systems, sufficient margin of control must be provided (*a*) to adjust for relatively slow changes in reactivity (reproduction ratio) due to burnout of the fuel; (*b*) to provide a margin of reactivity to offset the accumulation of fission products (some of which may be effective absorbers of neutrons); and (*c*) to provide for variations in reactivity due to thermal expansion or contraction of active or structural parts of the reactor.

Note that the tenth neutron used in control completed the accounting of all neutrons in the generation. Omitted from the diagram for simplicity were losses by absorption in structural parts of the pile, use of neutrons to produce radioactive isotopes for sale, neutrons causing fission of plutonium, and other possible reactions.

Having accomplished a controlled chain reaction, the heat liberated must be extracted as rapidly as it is produced in order to maintain the temperature of fuel and structural parts at limits established by the materials. About 80 per cent of the energy appears immediately at fission as kinetic energy of the two fission fragments which move in opposite directions at high velocity. In this example the fuel is part of a solid fuel bar so that these fragments collide immediately with adjacent atoms. Actually, the fragments move only a few thousandths of an inch before the kinetic energy is given up to adjacent atoms and appears there as increased atomic or molecular vibration which is heat. Therefore the majority of the fission energy is available only as heat energy and at the point of fission. The heat must then flow out of the fuel bar through any barrier or "canning material" which is used to prevent the fission products from contaminating the heat-transfer fluid and be transferred to the fluid.

The remaining 20 per cent of the energy is approximately 2.5 per cent in kinetic energy of neutrons released, 2 per cent in prompt gamma rays, 4.5 per cent in binding energy released when neutrons are absorbed, and 10 per cent in radiations from unstable fission-fragment nuclei as they decay at various rates to stable isotopes. Most of this energy appears sometime as heat in various parts of the reactor and shield.

Power Nuclear-Reactor Design Problems.

Chain Reaction. Here is where the design engineer must compromise parasitic losses of neutrons by absorption in materials he puts in the reactor with the desired structural or other properties of the materials. Throughout the later discussions of structure, heat-transfer fluid, and the like, the requirement of low absorption is ever-present. This factor, more than almost any other, forces the design engineer to examine the possible applications of a much wider variety of materials and alloys than he would have to do in any

usual heat-power equipment. He cannot afford to overlook the possibilities of relatively unknown members of the periodic table.

In addition to materials compromise between absorption and desired physical properties, he must compromise the most favorable theoretical arrangements for supercritical reactivity with such practical items as the following:

1. Flat surfaces compared to cylindrical or spherical surfaces.
2. Practical lattice arrangement of fuel bars or shapes to permit loading and unloading, opposed to an ideal arrangement for neutron economy.
3. Space and ducts for heat-transfer fluid reducing neutron economy.
4. Space and arrangement of control rods compromised with the uneven distribution of neutron flux which they may produce, interference with fuel bars, and the like.

All such compromises will affect the economy of neutrons and the investment in the "critical mass" of fissionable material put in the reactor, both of distinct economic significance.

Heat Transfer. Here is perhaps one of the most challenging fields for engineering advance. The tremendous power densities and temperatures of the bomb have been produced. Power densities and temperatures can be produced in a power reactor as high as the heat-transfer and materials engineers can handle. Furthermore, some features of the problem are favorable. The heat can be made available in a solid material for conduction and transfer by forced convection to a fluid under controlled surface conditions. Compared to an ordinary fuel-combustion boiler with heat transfer from combustion gases, good design should permit much higher heat fluxes with resultant economy in size and cost.

Any material in the reactor is subjected to a neutron flux of great density. Neutrons are sufficiently potent projectiles that they can cause many effects besides fission and transmutation which have been discussed. The designer must examine any material in the pile for its "radiation stability." Changes may occur in structural properties such as elastic limit, elongation, and the like; thermal and electrical conductivity; and—most important in fluids—chemical dissociation. Light nuclei are most effective in slowing down neutrons and these nuclei likewise recoil with highest velocity; consequently, one might expect that chemical compounds containing hydrogen, and to a lesser extent carbon, may be dissociated. The poor radiation stability of organic compounds rules out a large number of possible heat-transfer fluids. In general, pure elements will be most stable and, in particular, noble gases and liquid metals are favored. Water in addition to possible chemical dissociation has a great disadvantage in that it boils at unacceptable pressures and temperatures. It is unwise to allow a phase change in the reactor because of possible sudden changes in reactivity beyond the small permissible margin in control. Gas volumes and pressures necessary for high heat flux make liquid metals more favorable for high power density. Economy in recirculating-pump power also favors liquid metals in closed cycles.

There are obviously a great many possible thermodynamic cycles. Also there are various applications such as aircraft, naval craft, mobile power plants, and the like. Many have factors decisive in the selection of the primary fluid. Gases, even with open-cycle gas turbines, possibly in jets or rockets, may be best on some of these. However, the path which landbased electric plants will probably follow seems fairly definite, i.e., liquid metal to steam, or perhaps a three-fluid system. In all probability each loop of the cycle will be closed for reasons of economy and radiation hazard.

Industry has had sufficient experience with mercury, a fairly difficult, chemically poisonous metal, to know that it can master the problems of handling liquid metals. This experience has also taught us that the problems may not be easy of solution.

Structural Problems. A fundamental factor entering almost every detail of reactor design is reliability for the expected life of the reactor. Almost any structural part will become radioactive in some degree after the pile has

operated at significant power for some time. Not only will transmutations occur in the basic material but also in its impurities. Consequently, essentially no parts, mechanisms, or instruments from within the pile can be handled directly for repair, replacement, or modification. Unless the designer has made provision for replacement of some part using special remote-control tools and providing adequate shields for the operators, a failure of some part may well mean the end of that reactor.

The possible life of a reactor may be very long indeed. If adequate attention is paid to the "radiation stability" of materials, there is no part which is basically short-lived. The shield does not become saturated with products of neutron absorption, nor do other structural parts in degree sufficient to limit their life seriously. However, the degree of reliability required is very great.

Some of the significant structural or mechanical design problems arise in providing loading and unloading mechanisms for fuel packages and source-material packages. Both, when in the reactor, are intimately associated with the heat-transfer fluid. If high rates of heat transfer are provided, and they should be, the packages will doubtless be in direct contact with the fluid. This requires opening fluid channels and extracting fuel packages without loss of radioactive heat-transfer fluid, or exposure of operators to radiation from the pile or from "hot" discharged fuel. In a high-performance heat-transfer system, maintenance of extreme purity of the heat-transfer fluid may require that air, dirt, and the like be prevented from entering the fluid channels during refueling.

Obviously, it will be desirable to design to refuel major power plants without interrupting the continuous output of power. This will involve some major problems. Not only must the loss or contamination of the fluid be prevented but its continuity of flow must be maintained. Furthermore, refueling cannot be allowed suddenly to change the reactivity enough to exceed narrow control limits. Negligible sudden changes in reactivity may require that the fuel and source-material charge be divided into a great many small elements in many channels.

Such subdivision may also be dictated by the fact that fuel at different locations may be exposed to different intensities of neutron flux. If so, the rate of consumption will vary. The same may be true of the formation of new fuel in different locations in the source material. Therefore the charging and discharging mechanism should be capable of selecting and loading different channels with different frequency. Furthermore, such patterns should be flexible to provide for changes over years of operation dictated by variations in cost and production of different fuels.

The discharge end of the refueling operation has its own special problems. All motions or actions must be accomplished essentially by remote control, including the sorting of different elements. The elements must be placed in suitable shielded devices for either storage or transportation. Obviously, some care must be exercised to avoid accumulating a supercritical mass of fissionable material. In such an operation the engineer should anticipate jams due to malfunction of valves, parts, damaged fuel elements, and the like, and prepare suitable remote-control devices for locating and clearing up such difficulties.

Summary of Design Problems. The foregoing discussion is illustrative rather than complete. There are many other design factors, such as corrosion, which have been omitted. There are major components, such as the shield, which have not been covered. However, it is important in this introductory discussion to indicate some of the classes of problems characteristic of this field, some new to the engineering profession:

1. Radiation stability of materials.
2. Nuclear properties of materials, fission, scattering, slowing down, absorption.
3. Radiation protection for personnel.
4. Extreme reliability.

Conclusion.

In concluding it may be appropriate for the author to offer some estimates of timing in the development and expansion of atomic power for use as electricity by industry and consumers.

The first main problem is to produce a high-temperature, experimental, controlled nuclear reactor, with advanced design and materials so it is a major step toward a large-scale power reactor. The author estimates that such an experimental reactor may be developed, designed, and built, and that it may produce electric power in 3 to 6 years. The main limitations in this stage are technical problems. The outline of problems in this paper should make it apparent that a much earlier achievement of stated objectives is not feasible.

The limit on the next few steps to a prototype of a practical large-scale power reactor is also technical. The author estimates that such a prototype may produce, say, 100,000 Kw. of electricity in from 7 to 15 years' time.

The prototype of a plant suitable for installation and operation at various sites delivering large-scale electric power from atomic energy is the start of another phase where factors other than technical may control growth. However, technical improvement will have to continue to reduce the cost of atomic power so it will expand naturally in our national economy. Therefore it is appropriate to consider the economics for a moment before proceeding with the analysis of timing.

On the much debated question of economy, the author's opinion is simply that there appears to be no technical problem which seems likely to prevent atomic energy from becoming competitive with other fuels to the considerable benefit of the national economy. Whether it will or will not, however, depends upon many factors, such as, intensity of technical effort, technical modifications which may be found necessary to implement national or world security, and existence or absence of a workable international control of atomic energy. With such major questions as these yet to be resolved, there is little to be gained by speculating on such economic considerations as investment, fuel costs, and the like. It is well to recognize that atomic-power plants in 20 years will probably bear little resemblance to our current technology or even our expectations. Therefore a forecast of investment, operating, and fuel costs, to be realized 20 years hence, is likely to be more misleading than useful.

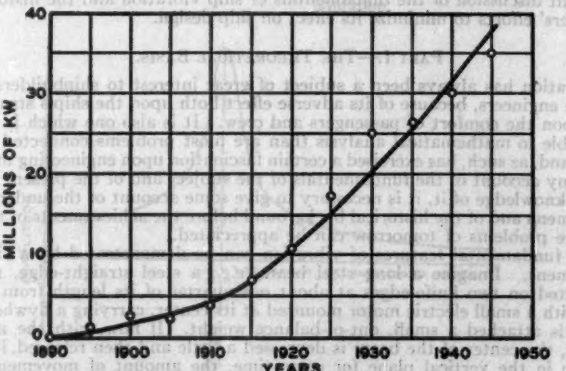


FIG. 3 INSTALLED CAPACITY OF PLANTS GENERATING ELECTRICAL POWER FROM FUEL IN THE UNITED STATES

If the growth of atomic power is controlled by economic factors as distinct from matters of national policy or security, its advent cannot be other than orderly. It will fit gradually and naturally into the pattern alongside coal, oil, gas, and water, increasing first in those areas which favor it economically. There is no basis for predicting any sudden dislocation or wiping out of any particular competitive source of energy.

Returning to the consideration of timing, factors additional to cost and technology affecting growth after a prototype are:

1. Expansion of atomic-power-plant manufacturing facilities.
2. Manufacture of ever-increasing quantities of atomic-power equipment. (The magnitude of this job is doubtless greater than the job of manufacturing steam boilers and turbines today.)
3. Installation and integration of plants by electric-utility operators in their power networks.
4. Training and expansion of technical and specialized personnel in both manufacturing and operating companies such as engineers, technicians, and health physicists for radiation safety control.

Similar factors to these were inherent in the growth of the present fuel-burning electric-power industry. Therefore Fig. 3, which shows this growth, is worthy of examination. A parallel relation to atomic-power expansion cannot be drawn too precisely. However, it may be safe to predict that power, comparable to our present capacity or a major part of it, may be achieved in 50 years. This is faster growth than the history of the steam-electric industry. It also seems clear that such a scale cannot be accomplished as soon as 20 years from now. To build, install, and operate an atomic-power industry comparable to our present steam-electric industry in 10 years would indeed mean taking maximum advantage of our great industrial capacity.

THE FUNDAMENTALS OF SHIP VIBRATION.

In order to obtain the high speeds required of many recent large commercial and naval vessels it has been necessary to develop high power at the propellers. The resulting excitation has introduced difficult and important vibration problems which have imposed definite limits on specific ship designs. In this paper, which appeared in three installments in "The Shipbuilder and Marine Engine-Builder" for May, June, and July, 1947, Dr. F. H. Todd gives an excellent discussion of the fundamentals of ship vibration and the history of engineers' efforts to minimize its effect on ship design.

PART I.—THE THEORETICAL BASIS.

Vibration has always been a subject of great interest to shipbuilders and marine engineers, because of its adverse effect both upon the ship's structure and upon the comfort of passengers and crew. It is also one which is more amenable to mathematical analysis than are most problems connected with ships, and, as such, has exercised a certain fascination upon engineering minds.

In any account of the fundamentals of the subject and of the present state of our knowledge of it, it is necessary to give some account of the underlying phenomena and of the historical background before the achievements of today and the problems of tomorrow can be appreciated.

The fundamental features of vibration can be demonstrated by a simple experiment. Imagine a long steel beam, *e.g.*, a steel straight-edge, to be supported on two knife-edges at about one-quarter of its length from each end, with a small electric motor mounted at its center, carrying a flywheel to which is attached a small, out-of-balance weight. If now, with the motor at rest, the center of the beam is depressed a little and then released, it will vibrate in the vertical plane for some time, the amount of movement, or "amplitude," gradually dying away. If the amplitude were recorded, the trace would be of the kind shown in Fig. 1; and, provided the initial displacement is not too large, the time interval between successive vibrations will be

practically constant. This interval is called the "periodic time," or "period," and the motion is said to be "isochronous." The number of vibrations per second or per minute is called a "natural frequency" of the system, and is obviously the reciprocal of the period of the motion.

If, now, the motor is run at any steady speed, the out-of-balance weight will transmit a so-called "periodic disturbing force" to the beam, which will then vibrate with a frequency equal to that of the impressed force—in this case, numerically equal to the revolutions of the motor. In effect, the beam is forced out of its natural frequency of vibration into that of the impressed disturbance, and such vibration is therefore referred to as "forced" vibration.

If the motor is run, successively, at a number of steady speeds, the amplitude of the vibration will be found to vary with the impressed frequency in the manner shown by the curve in Fig. 2. The chief feature is a marked peak in the amplitude which occurs when the frequency of the disturbing force is practically equal to the free, natural frequency of the beam. Such vibration is called "synchronous" or "resonant" vibration, and it is this type which causes most trouble in engineering structures and ships.

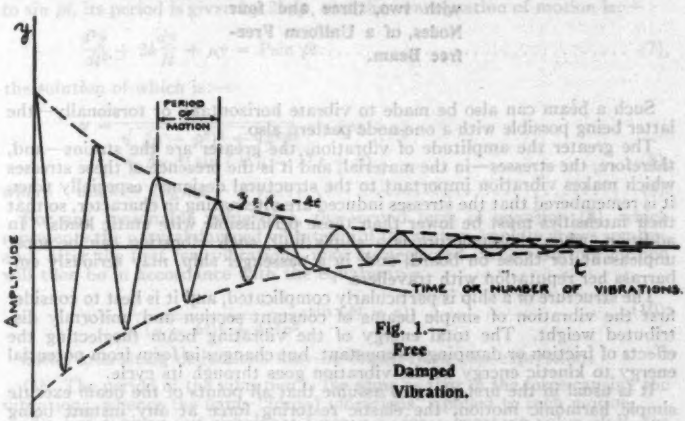


Fig. 1.—Free Damped Vibration.

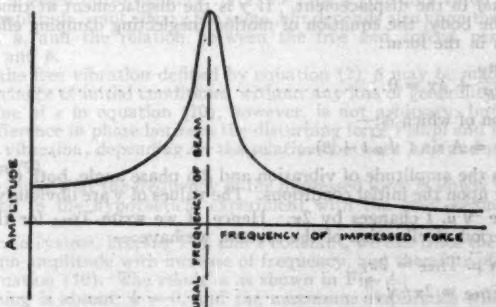


Fig. 2.—Variation of Amplitude with Frequency of Impressed Force

In the experiment envisaged in the foregoing, the beam remains at rest at the two points at which it is supported on the knife-edges; these points are called "nodes," and the beam is said to be executing vertical two-node vibrations. It can be made to vibrate vertically in other modes—with three, four or more nodes—the frequency increasing as the number of nodes increases. The shape of the corresponding amplitude curves or vibration profiles will be as shown in Fig 3.

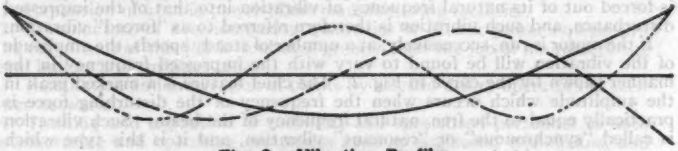


Fig. 3.—Vibration Profiles, with two, three and four Nodes, of a Uniform Free-free Beam.

Such a beam can also be made to vibrate horizontally or torsionally—the latter being possible with a one-node pattern also.

The greater the amplitude of vibration, the greater are the strains—and, therefore, the stresses—in the material, and it is the presence of these stresses which makes vibration important to the structural designer, especially when it is remembered that the stresses induced are alternating in character, so that their intensities must be lower than those permissible with static loads. In addition, when the structure is a ship's hull, such vibration can be most unpleasant for those on board, and, in a passenger ship, may seriously embarrass her reputation with travellers.

The structure of a ship is particularly complicated, and it is best to consider first the vibration of simple beams of constant section and uniformly distributed weight. The total energy of the vibrating beam (neglecting the effects of friction or damping) is constant, but changes in form from potential energy to kinetic energy as the vibration goes through its cycle.

It is usual in the first place to assume that all points of the beam execute simple harmonic motion, the elastic restoring force at any instant being proportional to the displacement. If y is the displacement at time t of any point of the body, the equation of motion (neglecting damping effects) may be written in the form:

$$\frac{d^2y}{dt^2} + \mu y = 0 \quad \ldots\ldots (1),$$

the solution of which is:—

$$y = A \sin (\sqrt{\mu}.t + \beta) \quad \ldots\ldots (2),$$

where A is the amplitude of vibration and β a phase angle, both of which are dependent upon the initial conditions. The values of y are obviously repeated every time $\sqrt{\mu}.\ t$ changes by 2π. Hence, if we write T_{free} for the free or natural period of vibration of the system, we have:—

$$\sqrt{\mu}.\ T_{free} = 2\pi,$$

$$\text{or} \quad T_{free} = 2\pi/\sqrt{\mu} \quad \ldots\ldots (3).$$

If the effects of damping forces, assumed proportional to the velocity at any instant, are included, the equation of motion becomes:—

$$\frac{d^2y}{dt^2} + 2k\frac{dy}{dt} + \mu y = 0 \quad \ldots\ldots (4)$$

the solution of this equation applies to the damped motion, and is of the form:—

$$y = Ae^{-kt} \sin (\sqrt{\mu - k^2}.t + \beta) \quad \ldots\ldots (5).$$

The sine factor oscillates between ± 1, so that the maximum amplitudes are given by Ae^{-kt}; and the amplitude now decreases with the passage of time, so that the vibration is gradually damped out. In fact, the equations $y = \pm Ae^{-kt}$ may be taken as those of the dotted curves bounding the vibration curve shown in Fig. 1.

The period of the motion is now given by:—

$$T_{damped} = 2\pi/\sqrt{\mu - k^2} \quad \ldots\ldots (6),$$

and is sometimes referred to as the "quasi-period." It is greater than that for the free, undamped condition; but since, for a given system, μ and k are constants, the motion is still isochronous.

When the system is subject to a periodic disturbing force, a further term has to be added to equation (4). If we assume this force to be proportional to sin pt, its period is given by $2\pi/p$, and the new equation of motion is:—

$$\frac{d^2y}{dt^2} + 2k\frac{dy}{dt} + \mu y = P \sin pt \quad \ldots\ldots (7),$$

the solution of which is:—

$$y = \frac{P}{\sqrt{(\mu - p^2)^2 + 4k^2p^2}} \cdot \sin (pt - \kappa) + Ae^{-kt} \sin (\sqrt{\mu - k^2}.t + \beta) \quad \ldots\ldots (8),$$

where $$\tan \kappa = 2kp/(\mu - p^2) \quad \ldots\ldots (9).$$

For any reasonable value of k, the second term of equation (8), which represents the natural damped vibrations of the system, rapidly grows smaller, and, after the first few vibrations, may be neglected. The resulting motion will then be in accordance with the equation:—

$$y = \frac{P}{\sqrt{(\mu - p^2)^2 + 4k^2p^2}} \cdot \sin (pt - \kappa) \quad \ldots\ldots (10).$$

Some interesting deductions may be drawn from this expression.

(1). The period of the vibration is the same as that of the force causing the vibrations, whence the name "forced vibrations" applied to such motion.

(2). For a given out-of-balance force, *i.e.*, for a constant value of P, the resulting amplitude is not always the same, but depends also upon the damping coefficient, k, and the relation between the free and forced periods, *i.e.*, between μ and p.

(3). In the free vibration defined by equation (2), β may be made zero by a suitable choice of initial conditions, without any loss of generality.

The value of κ in equation (10), however, is not arbitrary, but marks a definite difference in phase between the disturbing force P sin pt and the resulting forced vibration, depending on the relations between k, p and μ, as given in equation (9).

Suppose, now, that the frequency of the impressed force is slowly increased, as described in the hypothetical experiment with the steel beam and the motor.

For a given system, keeping P, μ and k constant, we can trace the variation of maximum amplitude with increase of frequency, and therefore of p, by the help of equation (10). The result is as shown in Fig. 4.

If damping is absent, $k = 0$, and the maximum amplitudes will be given by:—

$$y_{max} = P/(\mu - p^2),$$

and, for the particular value of $p = \sqrt{\mu}$, this becomes infinite. Now, this latter condition is equivalent to the statement that the period of the impressed

force is equal to the free, natural period of the system; and we have here the mathematical explanation of the condition of resonance, to which reference was made in discussing the beam experiments. In practice, of course, k is always finite, and, under such conditions, the amplitude at resonance will no longer be infinite. Since $(\mu - p^2)^2$ is always positive, it follows from equation (10) that the maximum amplitude will still occur when $\mu = p^2$, and will be given by:—

$$y_{max} = P/2kp.$$

Thus, the greater the value of k, the less is the amplitude at resonance; the latter is indeed very sensitive to the value of the damping coefficient, k, as may be seen from the typical results plotted in Fig. 5. It is also interesting to note that the maximum resonant amplitude occurs at the free period of the undamped system, and not at the quasi-period of the damped system.

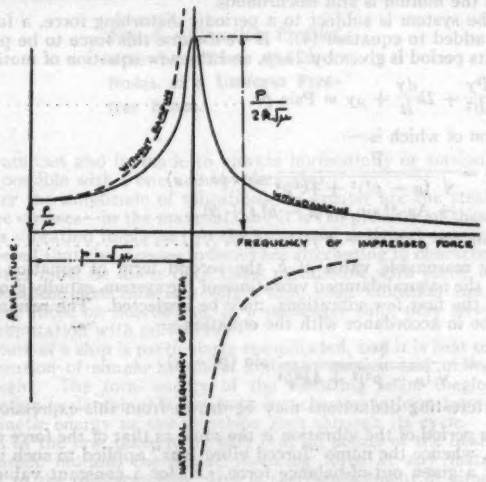


Fig. 4.—Forced Vibration and Resonance.

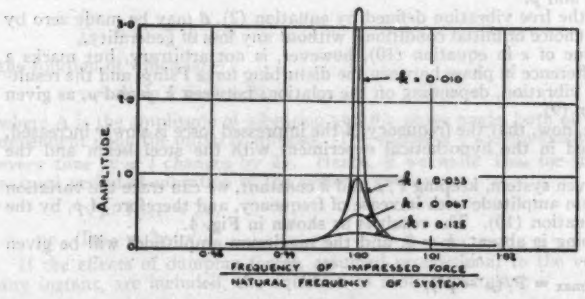


Fig. 5.—Effect of Damping on Resonance and Amplitude.

It may be concluded from this examination that, when a system is subjected to a periodic disturbing force, vibrations of excessive amplitude are to be expected when the period of this force is equal or close to a natural period of the system.

The forced vibrations which result from periodic disturbing forces are not, in general, in phase with the disturbance itself. The difference in phase is represented by κ in equations (9) and (10), and the result depends upon the damping coefficient k and the relationship between the natural and forced periods.

For the particular case of resonance, when $\mu = p^2$, we see from equation (9) that tan κ is infinite and κ is $\pi/2$, which shows that the actual vibration lags behind the force causing it by a quarter of a period, the relationship being exhibited in Fig. 6. The direction of the disturbing force is the same as that of the motion of the system, so that it tends continually to increase the amplitude.

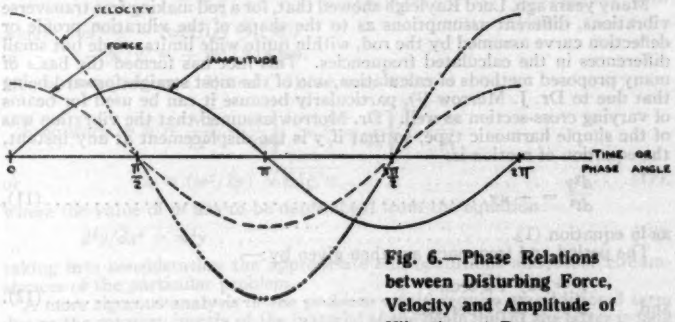


Fig. 6.—Phase Relations between Disturbing Force, Velocity and Amplitude of Vibration at Resonance.

A periodic disturbing force or couple may be resolved into harmonic components, the frequencies of which are usually 2, 3, 4, etc., times that of the primary component. Hence, a periodic disturbance having a frequency of $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{4}$, etc., of the natural frequency may become dangerous through one of its harmonic components. Where possible, structures subject to such forces should have their natural frequencies much below the running speeds. In such a case, there is no danger that the frequency of a harmonic component of a disturbance, arising from the running, will approach the natural frequency. In starting and increasing speed on such a machine, the resonant condition exists for such a short time that no excessive stress results.

In ship work, the foregoing is rather a counsel of perfection. In the first place, disturbing forces and couples, having a number of different frequencies, arise from the operation of the main and auxiliary engines and the propellers. On the other hand, the hull has a large number of natural frequencies—vertical, horizontal and torsional—each with two, three or more nodes. It so happens that, in a ship, the lowest natural frequency of all, *viz*, that of the vertical, two-node vibration, is usually not far, if at all, below the range of propeller revolutions commonly met with today; so that primary forces having a frequency of one per revolution are, in practice, very liable to excite the two-node vertical frequency of the hull, and, less frequently, the two-node horizontal critical. Higher harmonics, whose frequencies are equal to the Rpm. times the number of cylinders, or to the Rpm. times the number of propeller blades, may excite any of the higher modes.

It will be realized that the possibilities of vibration, from one cause or another, are very high, and it is therefore most necessary that the naval architect should be able to explore the whole problem in the design stage.

If the probable natural frequencies of the hull can be predicted with reasonable certainty, then due care and attention can be paid to the choice of the Rpm. of the main engines and auxiliaries, and to the balancing of these units, in order either to avoid resonance entirely, or to ensure that the residual out-of-balance forces and couples are so small that they do not cause vibration of measurable amplitude.

Hence, one of the primary problems facing the naval architect is to develop a method for calculating the critical frequencies of the hull from information available before the design begins to be expressed in steel, after which little can be done to alter the natural frequencies. The starting point is obviously a consideration of the more simple problem of the transverse vibrations of beams of constant cross-section, either uniformly loaded or carrying a single concentrated load.

Beams of Uniform Section.

Many years ago, Lord Rayleigh showed that, for a rod making free transverse vibrations, different assumptions as to the shape of the vibration profile or deflection curve assumed by the rod, within quite wide limits, made but small differences in the calculated frequencies. This fact has formed the basis of many proposed methods of calculation, one of the most straightforward being that due to Dr. J. Morrow (1), particularly because it can be used for beams of varying cross-section as well. Dr. Morrow assumed that the vibration was of the simple harmonic type, so that if y is the displacement at any instant, the equation of motion is:—

$$\frac{d^2y}{dt^2} = -\mu y \qquad \ldots\ldots(11),$$

as in equation (1).

The period and frequency are then given by:—

$$\left.\begin{aligned} T &= 2\pi/\sqrt{\mu}\ \text{sec.}, \\ \text{and}\quad n &= \sqrt{\mu}/2\pi\ \text{per sec.} \end{aligned}\right\} \qquad \ldots\ldots(12).$$

Hence, we can write:—

$$\begin{aligned} \mu &= -\frac{d^2y}{dt^2}/y \\ &= -\ddot{y}/y \qquad \ldots\ldots(13) \\ &= -\frac{\text{acceleration}}{\text{displacement}} \end{aligned}$$

Now, consider a beam uniformly loaded (*e.g.*, under its own weight) and of constant cross-section. Let the weight per unit length be w.

If the deflection at any instant at a point distant x from one end be y, the force due to a length of beam, δx, at this point is given by the product of its mass and acceleration, *i.e.*, by $(w/g)\,\ddot{y}\,\delta x$, and the force per unit length of beam is $(w/g)\,\ddot{y}$.

The ordinary theory of bending for beams leads to the equation:—

$$\text{Load per unit length of beam} = EId^4y/dx^4,$$

where E = modulus of elasticity of the material of the beam,

and I = the moment of inertia of its cross-section about the neutral axis normal to the plane of vibration.

Now, in the present problem, if we assume the beam to vibrate about its static-deflection or equilibrium position as the mean position, the additional load per unit length is that due to the dynamic forces associated with the vibration. Hence, we may write:—

$$\begin{aligned} &\text{Dynamic force per unit length of beam} \\ &= EId^4y/dx^4 \\ &= -(w/g)\,\ddot{y} \qquad \ldots\ldots(14). \end{aligned}$$

(1) Vibration of Beams of Irregular Section; J. Morrow; Phil. Mag., 1905.

If y_1 be the maximum deflection at the center of the span (y being that at a distance x from one end) the frequency at the center is proportional to $\sqrt{\ddot{y}_1/y_1}$; and, at a distance x from the end, it is proportional to $\sqrt{\ddot{y}/y}$. If the frequency is constant, the bar vibrating isochronously, we may write:—

$$\ddot{y}/y = \ddot{y}_1/y_1 = \text{constant},$$

since the frequency at every point in the bar is necessarily the same. Hence,

$$\ddot{y} = \frac{\ddot{y}_1}{y_1}\, y \qquad \ldots\ldots(15).$$

Equation (14) may now be written in the form:—

$$EId^4y/dx^4 = -(w/g)(\ddot{y}_1/y_1)\, y,$$

or,
$$d^4y/dx^4 = -(w/EIg)(\ddot{y}_1/y_1)\, y = m^4y \qquad \ldots\ldots(16),$$

where $m^4 = -(w/EIg)(\ddot{y}_1/y_1)$,

or $-\ddot{y}_1/y_1 = +m^4(EIg/w)$.

Hence, by equations (12) and (13), we may write:—

$$\text{Frequency} = n = \frac{\sqrt{\mu}}{2\pi} = \frac{1}{2\pi}\sqrt{-\frac{\ddot{y}}{y}} = \frac{1}{2\pi}\sqrt{-\frac{\ddot{y}_1}{y_1}}$$

or
$$n = (m^2/2\pi)\sqrt{EIg/w} \qquad \ldots\ldots(17),$$

where the value of m has to be determined from the equation:—

$$d^4y/dx^4 = m^4y \qquad \ldots\ldots(18),$$

taking into consideration the appropriate end conditions and other circumstances of the particular problem.

A more rigorous analysis of the problem would require an additional term due to the rotatory inertia of the material of the beam, but if the latter is long in comparison with its transverse dimensions, this effect may be neglected.

The solution of the differential equation (18) is a well-known standard form, *viz.*,

$$y = A\cos mx + B\sin mx + H\cosh mx + K\sinh mx \qquad \ldots\ldots(19),$$

containing four arbitrary constants, A, B, H and K, which must be determined by the end conditions of the beam.

The closest approach to a ship hull, vibrating when afloat, is a beam free at both ends and supported only at the nodes, the supports exercising no restraint on the movement of the bar; this is termed a "free-free" beam. The end conditions are that both the shearing force and bending moment at the two free ends must be zero. If l is the length of the beam, these conditions lead to the solution:—

$$\cos ml.\ \cosh ml = 1 \qquad \ldots\ldots(20);$$

whence
$$ml = 0,\ 4{\cdot}73,\ 7{\cdot}85,\ 10{\cdot}99,\ 14{\cdot}14, \text{ etc.} \qquad \ldots\ldots(21).$$

The frequency is given by equation (17), *viz.*,

$$n = (m^2/2\pi)\sqrt{EIg/w} = [(ml)^2/2\pi l^2]\sqrt{EIg/w} \qquad \ldots\ldots(22),$$

where ml may take in turn all the numerical values $\ldots\ldots(21)$.

Since the frequency depends on $(ml)^2$, the successive values of n will be in the ratios of the following numbers:—

$$4{\cdot}73^2,\ 7{\cdot}85^2,\ 10{\cdot}99^2,\ 14{\cdot}14^2, \text{ etc.}$$

or
$$1,\ 2{\cdot}756,\ 5{\cdot}404,\ 8{\cdot}933, \text{ etc.} \qquad \ldots\ldots(23).$$

We can now determine y for any particular value of ml in association with

the constants appropriate to the end conditions, and thus obtain the deflection curve or vibration profile for that particular mode of vibration. This will also give the nodes, or points of zero displacement. The first three values of *ml* lead to the results set out in Table I.

TABLE I.—CALCULATED POSITIONS OF NODES FOR A FREE-FREE BEAM.

Mode of Vibration.	Values of *ml*.	Position of nodes from one end of beam as fraction of length of beam.
1st	4·730	0·2242 0·7758
2nd	7·850	0·1321 0·5000 0·8679
3rd	10·996	0·0944 0·3558 0·6442 0·9056

We thus see that the first three values of *ml* correspond to vibrations with two, three and four nodes in the length respectively, and higher modes could be found by further calculation.

BEAMS OF VARIABLE SECTION.

When the loading and cross-section of a beam both vary along its length, the calculation of the periods by ordinary methods of analysis often presents great difficulty. Morrow's method can, however, be used with considerable success in such cases, since it can be applied to such problems by using graphical methods of integration.

Two curves, one showing the distribution of load, and the other the variation of the moment of inertia of the beam sections along the length, must first be prepared. A particular vibration profile, which fulfils, so far as possible, the known end conditions, must next be selected—for a ship it is usual to take that for a uniform free-free beam. Since the vibration is assumed to be simple harmonic in character, the ordinates of this profile curve will represent, though to two different scales, both the profile and the acceleration at any instant. If, therefore, the ordinates of the load curve and of the assumed profile at any particular point are multiplied together, the result will be the dynamic force acting at this point, *i.e.*, the product will give the value of the right-hand side of the equation (14) at that particular point in the length.

The curve of dynamic load, constructed in this way, can then be integrated graphically four times to give the final vibration profile, *i.e.*, a curve of y on a base of x. This must now be compared with the assumed profile, and, according to Rayleigh's principle, if they do not differ materially, the frequency will be almost the same for either profile, and it can be calculated in the way described earlier. Morrow's method, however, goes somewhat further than merely depending upon the correctness of Rayleigh's assumption, since it allows of successive approximation being made should the first derived profile differ much from that assumed—the calculation being repeated using the newly derived profile, until satisfactory agreement is obtained. The method also gives a wider range of choice for the shape of the vibrating bar. As an example, the frequency of a free-free uniform bar can be calculated without prior knowledge of the correct profile, found earlier by other methods. The end conditions, in this case, can be summarized as follows:—

$y = y_1$, the maximum deflection;
$d^2y/dx^2 = 0$ (d^2y/dx^2 is proportional to the bending moment),
$d^3y/dx^3 = 0$ (d^3y/dx^3 is proportional to the shearing force),

for both $x = 0$ and $x = l$.

In addition, the total external force on the beam must be zero, so that:—

$$\frac{w}{g}\int_0^l \ddot{y}dx = 0,$$

i.e., $$(w/g)\,(\ddot{y}_1/y_1)\int_0^l ydx = 0.$$

There are thus seven conditions, entailing seven arbitrary constants in the expression for y, and it may therefore be written in the form:—

$$y = A + Bx + Cx^2 + Dx^3 + Ex^4 + Fx^5 + Gx^6.$$

The frequency obtained from this assumption is

$$n = \frac{4 \cdot 732^2}{2\pi l^2}\sqrt{\frac{EIg}{w}}.$$

The constant, $4 \cdot 732^2$, compares with $4 \cdot 730^2$ obtained previously, so that, in this particular case, Morrow's method gives a result correct to the third figure at a first approximation. With beams of variable loading and cross-section, such accuracy is not to be expected from the first assumed profile, but usually a sufficiently correct estimate is obtained in these cases at the third approximation.

Lord Rayleigh showed further that a good approximation to the frequency of a beam could be obtained by the use of energy methods in association with any assumed and reasonable shape for the vibration profile.

We can represent the vibration by an expression of the type:—

$$y = u \sin pt \qquad (24),$$

where u is a function of x, which defines the shape of the vibrating beam. By the principle of the conservation of energy, we can write:—

Potential energy in extreme position = Kinetic energy in mean position.

At any given point on the beam, the maximum value of y is u, and the maximum velocity is pu. The bending moment on the beam (at the same point) corresponding to the maximum amplitude, *i.e.*, to the extreme position, is EId^2u/dx^2, so that the potential energy of the whole beam, which, as usual

is given by $\frac{1}{2}\int_0^l (M^2/EI)\, dx$, becomes $\frac{1}{2}\int_0^l EI(d^2u/dx^2)^2\, dx.$

The kinetic energy in the mean position, when the velocity is a maximum, is:—

$$\int_0^l (wv^2/2g)dx, \text{ or } \frac{1}{2}\int_0^l (w/g)\, p^2u^2\, dx.$$

Equating these energies, we find:—

$$p^2\int_0^l (w/g)\, u^2\, dx = \int_0^l EI\, (d^2u/dx^2)^2\, dx \qquad (25).$$

When the form of the function u is known, the period can be found. The useful thing about this result is that the value of p obtained is not affected very much by the form of u so long as it resembles reasonably the vibration profile. To show this, consider a uniform beam clamped at both ends. The end conditions in this particular case lead to the same solution as those for a free-free beam, so that ml in equation (22) is $4 \cdot 73$.

If we assume the beam to take the same form when vibrating as it would assume, statically, under the given loading, then:—

$$u = \pm x^2 (l - x)^2,$$

and, substituting this in equation (25), we are led to the value $4 \cdot 739$ for ml.

Alternatively, assuming that

$$u = 1 - \cos (2\pi x/l),$$

we find $ml = 4 \cdot 774$—not such good agreement as from the earlier assumed form of u, but still not very far different. In general, reliable results are obtained if the static deflection curve under the given loading is used for u in equation (25).

A ship's hull is an elastic girder of box-like section, and possesses natural frequencies of all the types mentioned in the foregoing, so that, if it is subjected to periodic disturbing forces which approach any of its natural frequencies, resonant vibration of uncomfortable amplitude is likely to result. Such disturbing forces may arise from the main engines, auxiliary machinery or propellers; and, while the first-named two can be eliminated or reduced to acceptable amounts, propeller forces always exist in greater or lesser degree. Once a ship has been built, it is practically impossible to change the natural frequencies of the hull, and resonant hull vibration, if it occurs, can be eliminated only by changes in engine balance or by altering the running speed of the offending machinery.

As a sequel to the foregoing description of the underlying physical phenomena, it will be of interest to trace the methods whereby the problem of ship vibration has been attacked by naval architects in the past, and then to pass on to an appreciation of our present knowledge and the methods available today for dealing with this troublesome problem. These aspects form the subjects of succeeding parts of the present article.

Part II.—Historical Survey.

In the latter part of the nineteenth century, there was a rapid increase in the speeds of new ships and in the powers of their machinery installations, and many cases of serious vibration were encountered, especially in torpedo-boats, which, in those days, were of very light construction and were fitted with reciprocating steam engines. At first, this vibration was regarded as a necessary result of the higher powers, and little was done towards avoiding it, recourse being had to local stiffening to eliminate or reduce unpleasant local vibration. At the same time, the engineer tended to lay the blame at the shipbuilder's door, on the plea that the ships' hulls were of too light scantlings. From what has been said in the first section of this article, it will be clear that this accusation could not have been generally true, since, as we have seen, any elastic structure will suffer from serious vibration if resonant conditions occur. Making the structure stiffer will not eliminate such vibration, but will alter the natural frequency, and so move the resonant range along the frequency base.

The first systematic investigation of ship-hull vibration seems to have been undertaken by Herr Otto Schlick who, in 1884, published the first of a long series of papers on this subject. He there pointed out that when there is a general state of vibration in a vessel's hull, it is not due to any weakness in the structure, nor to excessive engine power, but rather to resonance between some periodic out-of-balance force in the engine and one of the natural frequencies of the hull girder.

We have seen that, for a simple free-free prismatic bar, of constant cross-section and weight per unit length, the natural frequencies are given by:—

$$n = \frac{(ml)^2}{2\pi l^2}\sqrt{EIg/w} \qquad \ldots\ldots(22)\ (bis),$$

where ml is a numerical factor depending upon the mode of vibration of the beam. The last equation may be written in the form:—

$$n = \frac{(ml)^2}{2\pi}\sqrt{\frac{EIg}{wl^4}} = \frac{(ml)^2}{2\pi}\sqrt{\frac{EIg}{Wl^3}} \qquad \ldots\ldots(26),$$

where $W = wl$ = total weight of the bar, and n is the frequency per second.

Schlick applied this formula to ships by using a coefficient ϕ such that:—

$$N = \phi\sqrt{\frac{I}{\Delta L^3}} \qquad \ldots\ldots(27),$$

where

N = frequency per minute;
I = moment of inertia, in in.^{3}ft.^{2} units, of all longitudinal material in the cross-section amidships;
Δ = displacement in tons;
L = length in feet; and
ϕ = a coefficient, incorporating the constant factors from equation (26), and also an empirical term taking account of the different modes of vibration and of departures of the ship girder and its loading from the basic, uniform bar.

Average values of ϕ for the lowest, or two-node, vertical frequency of the hulls of different ship types were given by Schlick as follows:—

ϕ = 157,000 for torpedo-boats;
ϕ = 143,500 for mail steamships;
ϕ = 128,000 for cargo steamships.

On account of the wide variations which may occur among ships of any one of these classes, the Schlick coefficient can be used satisfactorily only if it has been determined experimentally for a previous ship of very closely similar size and arrangement.

Schlick considered that the transverse reaction of the propeller, the inertia forces arising from the reciprocating masses of the engines, and forces set up by shafts being out of alignment were the chief causes of such vibrations in steamships. The forces set up by these factors are periodic, their frequency being equal to the Rpm. of the engines, or some multiple thereof.

The section of a ship offers considerably less resistance to flexure in the vertical plane than to horizontal flexure, so that the natural frequencies of vertical vibrations are lower than those of horizontal vibrations for the same mode; and, for a given disturbing force, they are of larger amplitude. To quote from Schlick's paper, "the theory of synchronism is borne out by actual observations. In ships which exhibit strong vibrations, these are set up suddenly, and often with almost a shock, as a certain number of revolutions is attained." In the same paper he also discussed a case of horizontal vibration, which he was inclined to attribute to torsional vibration of the hull.

At the same time, Sir Alfred Yarrow had been working on the marine engineering aspect of the problem, and, in 1892, he presented a paper to the Institution of Naval Architects, in which he described a vibrograph for the automatic recording of vibrations, the principle employed being exactly the same as that employed in modern instruments. The essentials were a heavily-weighted platform, on which was mounted a vertical drum, driven by clockwork, and carrying recording paper, the whole platform being elastically suspended from a main framework which carried a pen to mark the paper. When placed on a vibrating structure, the framework and pen partook of the vibratory motion, while the drum, due to its great inertia and elastic supports, remained relatively at rest in space, and the pen recorded the vibration of the vessel.

Yarrow expressed the opinion that vibrations were caused by the inertia forces of the reciprocating masses in the engines, except that they might also be due to bad workmanship or want of repair, *e.g.*, screws not being concentric with the shafts, or the area and pitch of the propeller blades not being identical. He demonstrated this by running the engines of a torpedo-boat, both with and without propellers, and showed that, at the same Rpm., the vibration was identical in both cases, thus proving that vibration could be caused independently of bad workmanship in shafts and propellers, if the engines were not correctly balanced.

The reciprocating engines of those early torpedo-boats ran at very high Rpm., and Yarrow drew attention to the possibility of vibration at a number of resonant speeds. These he determined by running the engines, without propellers attached, in smooth water, the resonant speeds and the number of nodes in the length being clearly demonstrated by the ripples created on the

surface of the water. He instanced one such vessel as having shown severe vibration at 200, 400, 600 and 800 Rpm. These speeds represented successive higher frequencies, corresponding to the different values of *ml* in equation (22), and each related to a different mode of vibration, the number of nodes in the length having been two, three, four and five, respectively.

Sir William White, in 1892, referred to vibration in some torpedo gunboats. Careful and extensive observations were made on one of these vessels, and demonstrated that, however unpleasant the vibration might feel, it was of very small absolute value, the chief cause of the vibrations having undoubtedly been the forces due to the reciprocating masses in the engines. When the revolutions approximated to one of the natural frequencies, the vibration attained its maximum value, and practically disappeared at other Rpm. In the particular vessel tested, the speeds at which vibration was produced were approximately 65, 130, 190 and 260 Rpm. The amplitude was greater at the lower speeds, contradicting the view that the vibrations were due either to weak hull construction or to too powerful engines. In two vessels of the class, indeed, extra keelsons and deck stiffening were introduced, without effect upon the vibration phenomena.

In 1893, Schlick described an instrument he had devised for continuously recording vibrations in both the vertical and horizontal planes, and which he called a pallograph. Like the Yarrow instrument, it depended upon the inertia of two heavy masses, one free to swing in the vertical plane, the other in the horizontal. Schlick used this instrument on the German twin-screw despatch vessel *Meteor*, and found the maximum vibrations in the vertical and horizontal directions to be near the stern, with amplitudes of $\frac{3}{8}$ in. and $\frac{1}{8}$ in. respectively. Both the vertical and horizontal vibrations increased and decreased periodically, the horizontal vibration having reached a minimum when the vertical vibration was a maximum, and *vice versa*, as might have been expected from the reaction between the two engines. This "beating" phenomenon is common to all multi-screw ships. As before, Schlick maintained that the smaller vibrations were not truly horizontal, but were components of a torsional vibration, supporting the contention with records taken at various points on a cross-section of the ship.

Both Schlick and Yarrow constructed models of engines which could be placed upon model beams, so that the influence of Rpm. and the position of the engine along the length could be determined experimentally. The outcome of these experiments was the introduction of the four-crank, triple-expansion engine, with an improved mode of balancing known as the Yarrow, Schlick and Tweedy system. This largely eliminated the unbalanced inertia forces, and greatly reduced the vibration of ships in which such engines were fitted. In still water, with no screws on the shafts, there was practically no vibration of the first order at all; but, at sea, with the propellers in operation, vibrations of higher order appeared, due to blade and shaft irregularities and to wake effects. Such vibration was of small amplitude, but its relatively high frequency made it sometimes very unpleasant.

In 1895, Schlick gave the results of some investigations he had made concerning the position of the nodes in vessels vibrating in their fundamental mode. In every case, he found that the nodes were nearer to amidships than given by theory, the forward node showing the greater divergence. This, of course, was due to the differences between a ship's hull and the uniform prismatic beam to which the theory applied.

In the same paper, Schlick gave a formula for the torsional frequency of a hull, involving the polar moment of inertia of the cross-sections and the speed of longitudinal waves in the ship's structure. Mallock, in discussing this paper, objected to the use of the latter factor in the expression for torsional frequency, since it would seem to necessitate the existence of a fundamental frequency far higher than any engine speed. He quoted from a report (made to the Admiralty in 1892) in which the observed frequencies of horizontal and vertical vibrations in certain ships were stated to be in the ratio of the square roots of the horizontal and vertical moments of inertia of the midship section,

and he inclined to the view that the vibrations were truly horizontal, and not part of torsional vibrations.

In 1901, Schlick, in a further paper, described pallograph records taken on the German liner *Deutschland*. Before the trial, the engines were run, in smooth water, without propellers coupled to the shafts. The engines were balanced on the Schlick system, and, even when they were run at the critical speed, no vibrations were felt. At sea, with propellers coupled, however, some vibration, of a frequency of one per revolution, developed; and, after elaborate experiments, Schlick traced this to the propellers, stating it to be due to one blade of each screw experiencing greater resistance than the others, because of some difference in area, shape of sections, or pitch. He therefore recommended that greater care should be taken in the manufacture of propeller blades, in order to avoid vibration, save power and lessen the risk of blade fracture. The liner *Deutschland* was 662 ft. 0 in. in length, the power of her reciprocating steam engines having been 37,000 Ihp. The vessel had a two-node vertical vibration, of a frequency of 67 per minute, the maximum amplitude observed having been $\frac{1}{16}$ in., which was considered satisfactory.

Schlick presented the last of his notable series of papers to the Institution of Naval Architects on the occasion of its Jubilee Meeting, in 1911, when he reviewed the then state of knowledge of the whole subject. It was an appropriate time to do so, for, with the advent of the turbine, a new phase of the vibration problem had been introduced. Turbine machinery gave rise to no unbalanced reciprocating forces, and was, therefore, unlikely to excite the lower modes of vibration; the propellers, however, ran at much higher speeds, and vibrations due to defects in the blades, and to their passage through the varying wake, while usually of small amplitude, were, nevertheless, of high frequency, and, therefore, likely to be distinctly unpleasant to passengers and crew. They called for greater care in propeller manufacture, correct mass-balancing of the screws, and the provision of greater clearances between the blade tips and the hull, and between the propeller blades and the *A*-brackets or bossings ahead of them.

Schlick had confined his work to the measurement of natural frequencies of hulls and to the determination of suitable values of his coefficient ϕ for different types of ships. An alternative method of dealing with the problem was proposed, in 1895, by Mallock, who suggested the use of model beams for the prediction of the hull frequencies of actual ships. If the model and ship are made of the same material, and the distribution of weight and cross-section inertia along the length are the same in ship and model, the weight being reduced in the ratio of (scale)^3 and the inertia in the ratio (scale)^4, then:—

$$\frac{\text{Frequency for ship}}{\text{Frequency for model}} = \frac{\text{Length of model.}}{\text{Length of ship}}$$

If the ship's hull and the model beam are of different materials, allowance must also be made for the effect of the different elastic moduli. Mallock proposed the use of a plank of uniform thickness, the breadth varying so as to be a replica of the curve of the moments of inertia of the cross-sections of the actual vessel, thus giving corresponding stiffness. The plank could then be loaded with lead to give the correct distribution of load, the lead being in very short lengths so as not to contribute to the stiffness. From the observed frequency of this plank, that of the ship could be deduced by allowing for the effect of scale and that of the different elastic moduli.

Mallock first tested a uniform plank, and found the nodes to agree exactly in position with those deduced from theory; he also found the three-node and two-node frequencies to be in the ratio 2·73 : 1, as compared with the theoretical figure of 2·756 : 1. A dynamic model of a cruiser gave nodes much further from the ends than those of a uniform bar, and the three-node frequency was only twice the two-node figure.

In discussing one of Schlick's papers, Mallock also gave the ratios of the different frequencies for bars of constant depth, but with a plan-form consisting of two wedges, tapering uniformly from the full width amidships to a

point at each end (Table II); and he suggested that, in ships, the ratios were much nearer to 1, 2, 3, 4, etc.—a fact borne out by the results already quoted from the investigations of Yarrow and White.

TABLE II.—FREQUENCY RATIOS GIVEN BY MALLOCK, 1894.

Mode of vibration.	Uniform bar.	Bar of uniform depth, diamond plan-form.	Ship hulls (approx.).
2 nodes	1·00	1·00	1
3 ,,	2·76	2·28	2
4 ,,	5·40	3·7	3
5 ,,	8·93	5·9	4

The idea of using dynamic models was pursued somewhat further by Professor J. B. Henderson at the Royal Naval College, Greenwich. Using steel beams in exactly the same way as had been suggested by Mallock, Professor Henderson made dynamic models of H.M. scout *Pathfinder* and the liner *Lusitania*. In the first case, the frequency estimated for the ship from the results of the model experiments was 127 per minute, the assumed modulus of elasticity, E, having been 12,900 tons per sq. in. for both bar and ship. The actual, observed frequency was 106 per minute, and it was proposed to modify the value of E for the ship material, so as to bring the results of the two experiments into agreement, the ship value of E necessary to achieve this being 9,930 tons per sq. in., *i.e.*, a reduction of 30 per cent. For the *Lusitania*, the reduction would have to be about 24 per cent., to a value of 10,400 tons per sq. in.

This method of reconciling the results neglects certain other important factors. The modulus, E, always occurs in association with the moment of inertia, I, of the cross-section of the vessel, and it is equally possible to bring the results into agreement by adopting a lower value of I—and with more reason. There is always some doubt as to the amount of material which really does play an effective part in the longitudinal strength of a vessel, particularly deck plating fairly far removed from any continuous longitudinal material, and there is also the question of the allowance to be made for rivet holes. On the other hand, there seems no obvious reason why the value of E should be substantially less for material incorporated in the hull structure than for the same material in the form of a test-piece. In point of fact, opinion today is that, for the small deflections associated with vibration, practically all the longitudinal material is effective—even that in deckhouses—and that E has practically the full value, discrepancies between frequencies estimated either by calculation or from dynamic models, and those actually measured on board ship, being due to a third and hitherto neglected factor, *viz.*, the effect of the surrounding fluid.

In 1922, Tobin proposed to calculate the hull frequencies from a detailed consideration of the distribution of weight and inertia along the length, using the energy method described previously in equation (25). He made no attempt, however, to take into account the effect of the surrounding water, and, to reconcile the resulting calculated frequencies with those actually observed, a value of E below the test-piece value must be adopted. Tobin assumed the vibration profile to be the same as that for a uniform free-free beam, and he relied upon Rayleigh's principle to cover any departure from this shape. This it will do for any reasonable variation of profile shape, but only if the nodes are very nearly in the correct longitudinal positions. In ships, with the heavier concentration of hull weight amidships, the nodes tend to be further from the ends, as Schlick had previously observed; while the assumption that the profile is that of a free-free beam implies that the center of gravity of the total weight executes vertical vibrations.

A method of overcoming these objections was devised by Nicholls in 1924.

He was primarily concerned with excessive vibration in a certain destroyer, but he included in his paper a general discussion of resonant hull vibration. He, too, based his methods on the energy principle, but attempted to take into account more items in the balance between potential and kinetic energies, as follows.

In the extreme position, the potential energy is made up of:—

I.—Bending energy;
II.—Shear energy; and
III.—Gravitational energy.

In the mean position, all the energy is kinetic, and consists of:—

IV.—K.E. due to vertical velocity;
V.—K.E. due to angular velocity; and
VI.—K.E. of the water.

If the selected profile is such that the center of gravity of the vessel remains at rest during the vibration, then item III may be neglected. Depending on the assumptions made as to which items are considered negligible, the energy equation may then be written in the forms:—

$$\text{I.} = \text{IV.} \qquad (a);$$
$$\text{I.} = \text{IV.} + \text{V.} \qquad (b);$$
$$\text{I.} + \text{II.} = \text{IV.} + \text{V.} + \text{VI.} \qquad (c).$$

The first of these is the same as equation (25); both (*a*) and (*b*) had been covered by previous methods, but, as we have seen, the results were not satisfactory. Nicholls therefore investigated the third alternative, (*c*). He began with the profile for a uniform free-free beam, and modified it, for any given case, by moving the base-line an amount given by $\frac{1}{\Delta}\int mudx$, thereby taking into account the fact that the center of gravity of the ship must remain at rest.

Equation (*a*), modified to take account of the base-line correction, gave approximately correct results with a very low E value, *viz.*, 10,000 tons per sq. in. A more reasonable value of E, *viz.*, 12,900 tons per sq. in., gave the same numerical values of the frequency as (*a*), if the base-line correction were omitted, though the latter was necessary in order to reconcile the assumed profiles with those obtained on ships.

Nicholls was thus led to an investigation of item VI, and his model experiments on beams vibrated mechanically under water showed that the additional inertia required to allow for the virtual mass of the water might be as much as 70 per cent of the displacement, and therefore a matter of great importance. In the destroyer with which he was particularly concerned, method (*c*) gave correct numerical results if E were taken to be 12,900 tons per sq. in., and the increase of mass 50 per cent.

In 1927, Lewis presented a most interesting paper on vibration of ships generally, and included a tabular method of carrying out the calculation, based upon the energy equation. He also referred to the effect of the surrounding water, but left its numerical value to be found by correlation of calculated and observed frequencies. In the same year, Taylor in this country showed how the method proposed by Morrow (described earlier in the present article) could be applied to ship problems; and he gave an approximate method of allowing for the effect of shear deflection, which depends essentially on the ratios of length to depth, and breadth to depth, of the hull. He also referred to virtual-mass effects, but gave no method of including them in the calculation, leaving their effect to be taken care of empirically.

It was now obvious that little further progress could be made until the virtual-mass effects were more fully understood, and the work initiated by Nicholls was extended by Moullin and Browne at Cambridge and by Lewis in the United States of America.

When a vibrating body is immersed in water, its natural frequency is considerably below that which it exhibits in air. The water surrounding it is

in continual motion, and this necessitates the expenditure of energy. The effect is as though the mass of the body were increased, the new total mass being called the "virtual" mass, and the added amount the "added virtual mass." The virtual-inertia coefficient is the ratio of the total virtual mass to the ordinary mass of the body.

For a cylinder of elliptical section, totally immersed in an infinite fluid, and moving at right-angles to its longitudinal axis, its length in the direction of this axis being unlimited, the kinetic energy of the surrounding fluid per unit length of cylinder is given by:—

$$T = \pi b^2 w v^2/2 g \quad \ldots\ldots (28),$$

where T = kinetic energy of fluid;
b = half-breadth of cylinder in a direction perpendicular to that of the motion;
w = weight per unit volume of fluid;
g = gravitational constant, and
v = velocity of motion.

If m denotes the virtual increase in mass, then we have:—

$m = \pi b^2 w$ per unit length of cylinder.

This result applies only to cylinders of elliptical section, and depends upon the assumption that the motion of the fluid is two-dimensional. For sections of other shapes, and for cylinders of finite length, in which the motion will no longer be two-dimensional, we may write:—

$$m = c\,\mathrm{J}\,\pi b^2 w \text{ per unit length} \quad \ldots\ldots (29),$$

where c may be called the inertia coefficient for the section, and J is a factor to allow for the difference between the kinetic energy of the surrounding fluid in three-dimensional motion and in the two-dimensional case. To apply these considerations to ships, it has been assumed that, for a vessel floating freely at the surface of the water, the added virtual mass will be one-half that for the double body consisting of the under-water body of the ship combined with its "image" and immersed in an infinite fluid, so that

$$m = \tfrac{1}{2} c\,\mathrm{J}\,\pi b^2 w \quad \ldots\ldots (30).$$

A very complete investigation of the virtual-inertia effect has been carried out in the Cambridge University Engineering Laboratory by Moullin, Browne & Perkins, who derived a formula similar to (29), and evaluated c for forms of rectangular section and of different breadth to draught ratios. The values of an experimental inertia coefficient, c' were obtained, and curves of c'/c for various values of the breadth to draught ratio were plotted, the value being from 0·83 to 0·86 for typical ship proportions. This is, in effect, a measure of the coefficient J of equation (30).

Using the method of conformal transformation, Lewis obtained values of c for a series of cylinders whose cross-section shapes approached quite closely those of typical ship sections, and he also determined values of J for ellipsoids of revolution of different length to breadth ratios, and vibrating at right-angles to their axes. Calculations of the effect of the entrained water on the frequency of rectangular beams, made upon the basis of Lewis' work, are in excellent agreement with the experimental results by Moullin.

The question was also investigated theoretically by Taylor, who applied the method of conformal transformation to determine the virtual inertia of a series of prismatic bodies, and indicated the application of the results to the problem of ship vibration.

The subject had now reached a stage at which it was essential to compare the observed criticals with those calculated in the light of all this new knowledge. The present author, as holder of an 1851 Exhibition, carried out a research into ship vibration in the years 1926 to 1928, in the course of which he measured critical frequencies on some 17 ships. These were first analyzed on the basis of Schlick's formula, and the values of ϕ obtained, showed considerable scattering. To avoid the time absorbed in calculating the moment

of inertia of the midship section, the author suggested a formula of the same type, *viz.*:—

$$N = \beta \sqrt{\frac{B\,D^3}{\Delta\,L^3}} \quad \text{...(31),}$$

where B = breadth of ship, in ft., and
D = depth of ship to strength deck, in ft.

Calculations showed that, for vessels of given type, the moment of inertia was proportional to B D^3, and that, for any such type, β was practically constant. The correlation obtained by the use of this very simple formula was just as good as that afforded by Schlick's more complicated expression, average values of β being as in Table III.

TABLE III.—VALUES OF β ($= N \sqrt{\Delta L^3 / B D^3}$).

Types of Ships.	Condition.	Value of β
Large oil-tank vessels	Loaded.	62,500
Small, trunk-deck oil-tank vessels	Loaded.	45,200
Cargo vessels	Average trial condition.	48,350

All these apply to vessels in deep water—the frequencies will be altered when the water is shallow.

Subsequently, the natural frequencies of these vessels were calculated from the detailed curves of load and inertia distribution. The method employed was that first suggested by Morrow and developed by Taylor, the shear-deflection correction due to the latter having been incorporated. Now, however, the added virtual mass was also included in the calculation, the coefficients for each section being estimated from Lewis' results, on the assumption that the coefficient for any particular section was the same as that for an infinitely long ship of that same section throughout. The added virtual mass, so calculated for each section, was then added to the weight curve at each point along the length, and the new load curve so obtained provided the starting point of the calculation. It was found that the virtual-inertia factor (*i.e.*, total virtual mass/displacement) was a linear function of the ratio of breadth to draught, (B/d), the relation being:—

$$\text{Virtual-inertia factor} = \tfrac{1}{3}\,(B/d) + 1{\cdot}2 \quad \text{.......................(32).}$$

The dispersion of the plotted points for a range of B/d from 2·0 to 0·0 was extremely small, and this formula can be used as an excellent check on the calculation for virtual mass. It cannot, however, be employed to calculate the added virtual mass and the latter merely added to the displacement; the correct distribution of the added mass is most important.

The calculated frequencies (using all continuous longitudinal material in assessing the moment of inertia, with a test-piece value of E of 30×10^6 lb. per sq. in.) were in extremely good agreement with those observed in the ships, any small differences having been, in general, capable of at least qualitative explanation, *e.g.*, different types of hull construction. Calculations for one particular ship showed that the calculated frequency was relatively insensitive to variation of I along the length, the difference resulting from the substitution of a constant value of I (equal to that amidships) for the actual curve of I being only some 2 per cent. The effect of weight distribution, on the other hand, was found to be most important, the difference in the calculated frequency corresponding to the correct distribution and a uniform distribution being 24 per cent.

The possibility of computing the frequency with reasonable accuracy by this method represented a great advance, since the calculation could be made during the design of the vessel; the method required no exercise of judgment

at all, nor did it involve the knowledge of any empirical coefficient derived from a closely similar ship. It also enabled the effect of changes in the total loading or its distribution to be calculated with considerable accuracy.

In 1934, Burrill suggested the use of a modified Schlick formula which included one factor to take into account the shear deflection and another to allow for the virtual inertia of the water, on the assumption that the latter was a function of the ratio B/d.

In 1935, the present author showed that the virtual-inertia coefficient for a section could be expressed as a function of the ratio of breadth to draught and the sectional-area coefficient for that section, expressed in terms of the breadth at the water-line and the draught, both measured at the particular section concerned. This advance enabled the virtual mass to be calculated much more quickly than before, and considerably simplified the work. Further results of measured and calculated frequencies of ships' hulls, mostly for passenger ships with long superstructures, were presented to the British Association in 1938, and it was shown that the detailed method of calculation again gave reliable results, provided that, in calculating the moment of inertia, all material in those superstructure decks which covered 60 per cent or more of the vessel's length were included.

A very comprehensive paper on vibration as it concerns the marine engineer was given, in 1939, by T. W. F. Brown, who described the improvement in rigidity of the main-engine structure and the consequent reduction in the forces and moments. Moreover, in Diesel engines, the increase in the number of cylinders with larger powers increased the possibilities of achieving almost perfect balance.

It cannot be claimed that the foregoing historical survey is either complete or exhaustive, for there have been many other contributors to our present knowledge and to the literature of vibration. An endeavor has, however, been made to give a connected account of the progress which has been made during the past 50 years, without digressing into the more technical by-ways which branch from the main road.

It remains to review the present position, and perhaps to suggest lines of further research likely to be most fruitful in promoting a better understanding of the problem and of greatest help to the ship designer and marine engine-builder.

Part III.—Present Knowledge of Ship Vibration, Its Prevention and Cure.

All hull vibration has its origin in the main or auxiliary machinery, shafting or propellers. These may give rise to periodic disturbing forces or couples having frequencies equal to the Rpm. or some multiple thereof. So far as the main and auxiliary engines are concerned, the forces arise from the reciprocating masses, if these are not completely balanced, and from the torque reaction of the engine upon its seating, and so upon the hull. The frequencies will depend upon the type of engine, upon the number of cylinders and method of balancing.

The reciprocating steam engine was a frequent cause of vibration until much greater care was devoted to the accurate balancing of its moving parts. The steam turbine itself gave little trouble, and interest in hull vibration was revived with the successful application of the heavy-oil engine to marine propulsion. This is not to say that the internal-combustion engine is inherently a worse offender than the reciprocating steam engine—such units may, in fact, be extremely well balanced—but it so happened that the Rpm. of the first marine Diesel engines were higher than those of steam engines of comparable power, and frequently synchronized with the two-node vertical hull critical. There were at that time several noteworthy instances of serious vertical vibration, and these, in turn, led to a great improvement in engine balancing. Today, the tendency in Diesel-engine design is towards still higher Rpm., because of the greater power-weight ratio so obtained, and it is

probable that the greatest danger now arises from synchronism with the two-node horizontal critical.

The propeller, too, may give rise to vibration, quite apart from the main engines. If the propeller as a whole is out of balance, *i.e.*, if its center of gravity is not on the shaft axis, there will be an out-of-balance force, in the transverse plane, of a frequency of one per revolution, and this is capable of exciting hull vibration by transmission through the shafting. Again, if the pitches of the blades are not all exactly the same, there will be a net resultant force; and, since the slip angles are invariably small, a small difference in pitch may give rise to a relatively large disturbing force. Though both these causes of vibration may be avoided by extreme care in propeller manufacture, there is a third which is always present. This originates in the propeller, and is due to the varying velocity field in which the blades operate during a single revolution. This variation is due to the wake created astern of the hull, and results in a periodic variation in the thrust and torque experienced by the blades, and hence in the reactions transmitted to the hull. While this cause of vibration can be reduced by careful attention to the design of the stern arrangements as a whole, it is always present to some extent; and, since it has a relatively high frequency (equal to the product of the Rpm. and number of blades), its effects may be distinctly unpleasant to passengers and crew, for such vibration may generate a great deal of noise from cabin fittings, doors and similar objects.

If the out-of-balance forces and couples are of small absolute magnitude, they will excite unpleasant vibration only if they are in synchronism with one of the natural frequencies of the hull or of some part of it. If they are large, they will cause forced vibration throughout the range of engine Rpm., with serious vibration in the resonant regions.

The Cure or Reduction of Existing Vibration.

Serious vibration in a new ship is almost invariably of the resonant type. It is always due to the presence of some unbalanced periodic force, and the most complete and satisfactory cure will be to reduce or eliminate that force. As a first step in this direction, vibrograph records should be taken; from these, the resonant frequency may be accurately determined, and the source of the vibration can then usually be detected. If it lies in a small auxiliary machine, it may be possible to alter the Rpm. so as to destroy synchronism, or to improve the balancing, or, again, to adopt an elastic seating which will absorb the energy of the disturbing force and so prevent its transmission to the hull.

When the main engine itself is the cause of the trouble, the remedy is not so easy. Here, again, the engine balance should first be studied to see whether it can be improved. For example, some engines are completely balanced in the vertical plane, but have a primary horizontal out-of-balance force; and, if the Rpm. in service should coincide with the two-node horizontal frequency, serious vibration would result. In such a case, the balance could be modified so as to eliminate the horizontal disturbing force; any vertical force introduced by the change would, normally, be acceptable, since it would cause vertical vibration of serious amount only in the neighborhood of the two-node vertical critical frequency, which, being considerably below the service Rpm., would occur only as a transient during the running-up or slowing-down of the engines.

Another means of curing such vibration is to reduce the service Rpm. below the critical range by increasing the pitch of the propeller; but this method is practicable only if the engine is sufficiently flexible, so that it can continue to deliver the required power at the lower revolutions. This is usually possible with turbines, but, with reciprocating steam or oil engines, it entails a corresponding increase in the mean effective pressure; and, if this is not acceptable, a loss of ship speed will result. The alternative of increasing the Rpm. above the critical value by decreasing the pitch of the propeller, though it avoids the difficulty associated with the mean effective pressure, necessitates higher

piston speeds; and, moreover, if the engine speed has to be reduced at sea, due to heavy weather, it may again come within the resonant range.

Elastic mountings for main engines are a delicate problem. Their frequency must be considerably lower than that of the disturbing force, and, with a heavy engine, this involves highly elastic springs, allowing a large movement of the engine itself, with all the consequent difficulties of pipe connections and the inertia forces due to acceleration during pitching and rolling. Incorrect design of such seatings may, in fact, increase the disturbing forces.

When the objectionable vibration is localized, the remedy is to be found in local stiffening; while elastic mountings for cabin and other fittings would do much to reduce the noise and irritation occasioned.

When the vibration has its origin in the propellers, and is of a frequency of one per revolution, the mass balance of the propeller and the pitch of all the blades should be checked. This may show which of these two factors is the cause, and may also indicate where the cure is to be found. If it is of the blade-frequency type, and resonant in character, it may be eliminated by the fitting of new propellers having a different number of blades; and, simultaneously, the opportunity should be taken to alter the rake of the blades, so as to increase the clearance between blade tips and hull, and between the leading edge of the blades and the body-post or shaft-web endings, thus reducing the wake variation over the disc.

Resonant hull vibration is not amenable to treatment by structural changes—to alter the natural frequency by 5 per cent involves an increase in the inertia of the midship section of 10 per cent, and it is generally impracticable to introduce sufficient material into an existing ship to achieve this.

In twin-screw ships, vibration of the beating type occurs due to the "phasing" of the two engines when they run at slightly different speeds, and this can sometimes be reduced by tying together the tops of the engines, so as to prevent their relative movement.

One further means of reducing vibration is the vibration damper or neutralizer. The principle of such devices was described by Professor C. E. Inglis in 1933. They consist essentially of a spring-supported mass having a frequency equal to that of the vibration, in association with a damping arrangement in the form of a dash-pot or friction plates. Inglis suggested a type suitable for use in cases of local vibration of deck beams, the mass being about 1 per cent of that of the girder. The same method has been applied to resonant vibration of the whole hull by Dr. Loser, in Italy. His neutralizer consists of a cast-iron tank, subdivided into small cells and filled with water to any desired level, and supported on a number of double springs. The arrangement must be firmly connected to the ship's structure, preferably at the point of maximum amplitude, which is usually the extreme aft end. The frequency of the neutralizer can be adjusted to cater for any particular vibration by alteration of the water-level inside it. In one instance, in a ship subject to strong vertical vibration, the neutralizer was successful in reducing the amplitude to only 6 per cent of its original value, the weight of the neutralizer having been 12 tons and the ship displacement 11,700 tons. This device would seem to have considerable advantages in connection with synchronous vibration caused by propeller blades, where, as we have seen, the presence of unbalanced forces is unavoidable.

The Prevention of Vibration.

The correct time to face the problem of ship vibration is in the very early design stages, when all the arrangements are still in a fluid condition, and not on the trial trip, when it may have become a serious matter, which can be treated only by one or more palliative measures.

All that has been said about the correct balancing of the main engines, auxiliary machinery and propellers applies equally to the design of a new ship. The wake variation over the screw disc must also be reduced to the minimum by meticulous attention to the details of the stern arrangements. In single-screw ships, to avoid eddying in front of the propeller, the after-end water-

lines must not be too steep, and the body-post should be shaped, while adequate clearances should be arranged in the aperture; also, the clearance between the trailing edges of the blades and the nose of the rudder or fin should be larger the greater the thickness of the latter. In twin-screw ships, the tip clearance should be as large as possible, consistent with the other considerations involved, and the endings of the shaft bossings should be well tapered and cut forward from the screw disc, in order to reduce, to the maximum possible extent, the belt of wake water leaving the webs. All these features are much more easily dealt with in a new design than in a completed ship.

When all details of this kind have been considered, there remains the possibility of resonant vibration of the local or general type. The former must, inevitably, be left to chance—it is physically impossible to calculate the natural frequency of every individual part of so complicated a structure as a ship—and any local vibration found in the finished ship must be dealt with by local stiffening, pillars and girders, or by the use of a vibration damper.

The possible occurrence of general resonant vibration is a much more serious matter. For ships with modern reciprocating engines, whether steam or Diesel, the most likely modes of excitation are the two-node vertical and horizontal vibrations, and every endeavor should be made to choose the Rpm. of the main engine in such a way that resonance shall not occur with either of them. This implies that the natural frequencies of the hull may be calculated while the design is still in its very early stages, and it has been shown that methods do exist which give satisfactory results. If information is available for a very similar ship, the use of a formula, *e.g.*, (27) or (31), will give a very close approximation. If such data are not available, the two-node vertical frequency may be calculated as described earlier, account being taken of the distribution of loading, inertia and entrained water.

The results plotted in Fig. 7 show that this method is capable of giving a quite close estimate of the ship frequency. No personal judgment need be

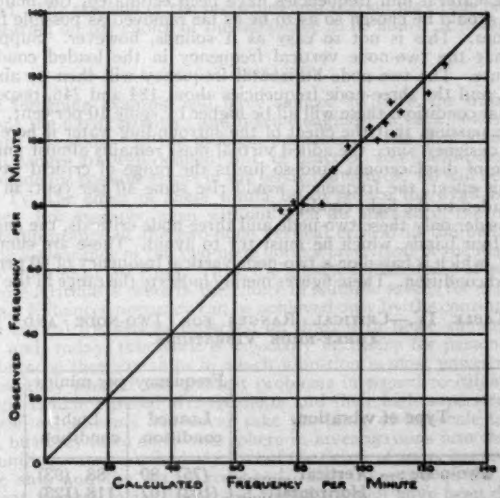


Fig. 7.—Comparison between Observed and Calculated Natural Frequencies for Two-node Vertical Vibrations of Ships.

exercised in the selection of an appropriate coefficient, nor is it necessary to have knowledge of a previous similar ship. It is unfortunate, however, that at present the greatest errors are liable to occur in connection with passenger ships, which are just those in which vibration is most unpopular with owners and passengers. This is due partly to lack of knowledge as to the extent to which erections above the strength deck contribute to the stiffness of the vibrating hull girder, and partly to the presence of the large open spaces required for engines, boilers and public rooms, and which makes it difficult to decide just which is continuous longitudinal material. The difficulty will slowly be overcome as more data become available for such vessels.

The two-node vertical critical can thus be estimated fairly closely, and we have confidence in the method of calculation, because it has been checked by the frequencies actually measured in a large number of ships. Much less is known of the corresponding horizontal frequency, and still less of the higher orders of both kinds, since they are of much smaller amplitude, of less frequent occurrence, and much more difficult to measure.

The two-node horizontal vibration is of a higher frequency than the corresponding vertical vibration, because the dimension of the hull girder in the direction of vibration is greater for horizontal than for vertical vibration, but the effect of entrained water in such a motion has not yet been investigated. From the data available, the ratio of the horizontal to the vertical two-node frequency is about 1·34.

For variations of higher orders, we have seen that the ratios for a uniform bar are 1·0, 2·76 and 5·40 for two, three and four nodes respectively. For a diamond-shaped beam of constant depth, these figures become 1·0, 2·26 and 3·7; and, if the correction due to shear is included, these ratios are further reduced. For very fine ships, such as destroyers, the successive criticals approach the ratios 1, 2, 3, 4. In only one instance has the author been able to measure the three-node vertical critical with certainty, and the ratio was then 2·3 for a large motor-driven passenger liner.

When the natural hull frequencies have been estimated, the main-engine revolutions should be chosen so as to be as far removed as possible from the critical values. This is not so easy as it sounds, however. Suppose, for example, that the two-node vertical frequency in the loaded condition is 80 per minute. The two-node horizontal frequency will then be about 107 per minute, and the three-node frequencies about 184 and 246, respectively. In the ballast condition, these will all be higher by some 10 per cent. It may be noted, in passing, that the effect of the surrounding water is here a great help to the designer, since the added virtual mass remains almost unchanged with change of displacement, and so limits the range of critical frequency. Without this effect, the frequency would rise some 40 per cent. in passing from the loaded to the light condition.

If we consider only these two-node and three-node criticals, the engineer is faced with four bands, which he must try to avoid. These are summarized in Table IV, which is based on a two-node vertical frequency of 80 per minute in the loaded condition. These figures merely indicate the range in the maxima

TABLE IV.—CRITICAL RANGES FOR TWO-NODE AND THREE-NODE VIBRATIONS.

Type of vibration.		Frequency per minute	
		Loaded condition.	Light condition.
Two-node :—	Vertical	(75) 80	88 (93)
	Horizontal	(102) 107	118 (123)
Three-node :—	Vertical	(176) 184	202 (210)
	Horizontal	(235) 246	270 (283)

of the vibration. To avoid unpleasant amplitudes, it would be necessary to keep the Rpm. of the main engine at least 5 per cent below or above the outside limits of the maxima (as indicated by the figures in brackets), thus broadening the bands considerably. In practice, a further broadening would be necessary to provide a margin against error in the estimated figures.

Moreover, it is not sufficient to consider merely the primary disturbing forces, since secondary unbalanced forces may be present in the engine system, and there will be higher frequencies from the propellers. Thus, if the engine speed is chosen at 95 Rpm., a three-bladed propeller would give a disturbance of a frequency of 285 per minute, *i.e.*, almost resonating with the natural three-node horizontal mode in Table IV. A further complication is that, today, it is usually necessary to adopt one of a number of standardized engines, and this, again, may lead to difficulty and compromise. Thus, if the most suitable engine for the vessel exemplified in Table IV were designed to run at, say, 120 Rpm., it would not be quite clear of the two-node horizontal critical range, and would be liable to resonate more sharply with it if the engine speed were to fall off because of bad weather or other reason, as it would certainly frequently do in service.

In such case, it would be necessary to balance out, as completely as possible, all primary horizontal forces, and, maybe, to accept some degree of primary lack of balance in the vertical direction. This would cause vertical vibration between 80 and 90 Rpm., but the effect would normally be experienced for only a few seconds as the engine ran through that range.

The longitudinal position of the engines also has an important bearing on the problem. The location of the machinery cannot be decided from considerations of vibration alone, but once it is fixed, and the Rpm. in service determined, its position relative to the nodes for the most dangerous type of vibration likely to be experienced should be examined. If the engine installation lies between two nodes, then unbalanced forces will be dangerous, while couples will not have any material effect in inducing vibration. If it lies across or near to a node, the conditions are reversed, and couples must be avoided. The position of the propelling installation may thus have an important bearing on the choice of the balancing to be aimed at in any given case

Conclusion.

The foregoing example will have sufficed to show the difficulties which confront the naval architect and marine engineer in their endeavors to avoid resonant vibration. In the problem described, it has been assumed that the critical ranges are well defined; but, in practice, some allowance would have to be made for possible errors in the predicted figures. As will be obvious from Table IV, the effect of these would be to reduce the available bands still further. For example, even without such an allowance, the range between the disappearance of the two-node vertical critical in the ballast condition, and the appearance of the two-node horizontal critical in the loaded condition is only from 93 to 102 Rpm. Thus, the greater the accuracy of our predictions, the more latitude is there in the choice of suitable speeds for engines and propellers. Enhanced accuracy can be achieved only by the continual correlation of the critical frequencies observed on ships with those subsequently calculated, and, today, this work is especially necessary for passenger ships, not only because they are ships in which vibration is most unwelcome, but also, because they present the greatest problems in regard to calculation, due to their particular internal arrangements and their high superstructures.

Model experiments can never take the place of full-scale tests on actual ships, but they have their own sphere in investigations into the effect of the surrounding water, particularly that of the depth of water between the bottom of the ship and the sea or river-bed. The literature of the subject bears eloquent witness to the labor and thought which have been devoted to the problem of ship vibration, almost since the days of the first steam-propelled ships. Progress has been stimulated, from time to time, by the adoption of various types of prime movers, and has been rapid of late years. It may be

that, with the coming of the gas turbine, the days of reciprocating marine engines are numbered, and this development will certainly remove some of the greatest problems in hull vibration; but, so long as ships are driven by propellers, those connected with the higher frequencies will remain.

We are also still very ignorant so far as the higher natural critical hull frequencies are concerned, and much remains to be done before all the problems of the future can be solved with certainty and confidence.

THE LAW OF SIMILITUDE FOR FLOW PROBLEMS, ITS EXPERIMENTAL VERIFICATION AND APPLICATION TO MECHANICAL ENGINEERING.

The laws of similitude are widely used in flow research. The following article, by Mr. Hablützel in the Sulzer Technical Review (No. 1, 1947), proceeds from a short description of their nature to the demonstration of their value for practical engineering, and particularly pump construction, on the basis of an example and the test results obtained with it.

Two flows are held to be similar when two geometrically similar fluid particles describe geometrically similar paths in geometrically similar boundaries. Furthermore, the forces acting upon the particles, *i.e.*, forces of pressure, inertia and friction, must bear the same relationship to each other in both cases. This requirement is in its turn fulfilled when the ratio can in both cases be reduced to the expression

$$\frac{\gamma \cdot w \cdot d}{g \cdot \mu} = \frac{w \cdot d}{\nu} = Re = \text{Reynolds number},$$

where γ is the specific gravity, w the velocity and μ the dynamic viscosity of the flowing medium, d the comparative length of the material boundary and g the acceleration due to gravity. The density $\frac{\gamma}{g} = \rho$ can be combined with μ to form the kinematic viscosity $\nu = \frac{\mu}{\rho} = \frac{\mu \cdot \gamma}{g}$.

With the aid of these simplifications the conditions for similar flow can be summed up in the correspondence of the Reynolds number *Re* and the geometric similarity of the material boundaries. The influence of gravity is neglected in these considerations, so that the conditions of similitude given do not hold good, or at least do not suffice, where gravity exerts any great influence on the flow in question, as for instance in the wave movements at a free surface or in cases where differences in the vertical extension of the flow sections under examination lead to considerable differences of pressure.

But if in similar flow, as stated at the outset, all the forces acting on the fluid particles are in the same ratio to each other and the paths described are geometrically similar, the components of work done by the forces in the actual movement must also be in the same ratio to each other. The negative work done by the frictional force is thus always proportional to the useful work of the other forces, so that the hydraulic efficiencies η_h are equal. It is of course to be remembered that the pressure forces are not determined by the absolute value of the pressure, but by the change of pressure in the direction of motion. If the absolute value of the pressure changes, the same η_h is only arrived at in similar flows when the pressure energy is in the same ratio to the velocity energy. This condition, as will be shown later, is fulfilled in geometrically similar flow machines with the same *Re*. Not only the hydraulic efficiencies, however, but also many other non-dimensional flow figures are identical in similar flow, e.g., the coefficient of pipe friction λ, as fixed by the equation of the resistance head

$$h_w = \lambda \frac{l \cdot c^2}{d \cdot 2g},$$

The identity of this λ can be practically established in simple experimental apparatus, and the theoretical grounds are likewise not far to seek. In flow through a pipe the force of inertia T is proportional to the square of the velocity c^2, for instance $T = k_1 \cdot c^2$; but the frictional force R is proportional to the magnitude λc^2, and thus for instance $R = k_2 \cdot \lambda \cdot c^2$. Consequently $\frac{T}{R} = \frac{k_1}{k_2 \cdot \lambda}$. In these expressions c is the mean velocity in the pipe cross-section, while k_1 and k_2 are functions of the distribution of the velocity over the cross-section and are accordingly equal for similar flows. As, however, the expressions $\frac{T}{R}$, k_1 and k_2 take on the same value for the same Re, this must also be the case for λ. The application of these principles revealed by Reynolds to practical experiments opens up a field of unexpected possibilities. The scale of models can be so chosen that the very large or very small dimensions of machine parts to be executed no longer represent any obstacle to testing, and in place of water, steam, etc., air for instance can be used, thus making observations and measurements possible which could not otherwise be envisaged. It is only necessary to see that the geometric similarity of the material boundaries and the correspondence of Re for model and execution are fulfilled. There can be no doubt that the immense progress made in recent times in the domain of aerodynamic machinery and aviation is in part due to the application of the laws of similitude to flow experiments.

The treatment of these problems in technical high schools and colleges is therefore a requirement of our time, and it is important that the student should not only be acquainted with the theories but should also have their correctness demonstrated to him by experiments in the laboratory, which will at the same time promote his confidence in modern experimental methods in general. For these reasons experiments have been carried out in the Cantonal Technical College at Winterthur, Switzerland, to determine the coefficent of pipe friction λ as a function of Re in a smooth, clean, drawn brass pipe of 35 mm. inside diameter and 2000 mm. length. The experiments were made with water and with air at roughly barometric pressure. The results of the repeated measurements are shown in Fig. 24 and confirm the law of similitude by a technically satisfactory correspondence of the test curves. With water, the velocity of which was about 0.5–1.5 m. per sec., the experiments offered no serious difficulties, and the flow velocity could be given very accurately by the weighing method. In the air experiments, on the other hand, measuring difficulties were encountered, for at air velocities of 6.6–17.2 m. per sec. the pressure losses for pipes 2 m. in length are already 4.4 to 22 kg. per sq. m. The determination of the weight of the flow with nozzles and diaphragms involves further sources of error. It is true that the flow figures at the cross-sections of measurement were established by calibration tests with water at similar values of Re, but it must nevertheless be stated that the accuracy of the measurements for water is greater than for air. As a matter of interest, the test curves are compared with a curve calculated by the empiric formula of Blasius, $\lambda = \frac{0.316}{\sqrt[4]{Re}}$, which holds good for smooth pipes at values of $Re < 10^5$. The divergences between the measured and calculated values are very slight and suggest at most that a pipe which is regarded as clean and smooth for technical purposes does not quite fulfill the requirements of Blasius.

The laws of similar flow can be carried over from single flow elements to flow machines as a whole, provided that the identity of Re is fulfilled as well as the geometric similarity. In this case, however, Re is defined by the relationship $Re = \frac{u_2 \cdot D_2}{\nu}$, where u_2 is the peripheral velocity at the impeller outlet and D_2 is the outside diameter of the impeller. This appears illogical in that u_2 is not a velocity of the flowing medium, but owing to the geometric similarity of the boundaries the velocity triangles are similar, so that all velocities of flow correspond to the impeller velocities. In all corresponding

sections of similar flow machines the values of Re therefore coincide when the expressions $\frac{u_2 \cdot D_2}{\nu}$ are equal and when the velocity triangles fixed by the blade angles reproduce the actual flow conditions. Similar velocity triangles, however, lead to the same degrees of reaction in the flow machines, or in other words to the same ratio of pressure energy to kinetic energy, which, together with equal values of Re, also results in the same hydraulic efficiencies η_h. According to Euler's theory the delivery head $H = \frac{\Delta p}{\gamma}$ depends only on the angles, on η_h and u_2^2, and if it is divided by the velocity head $\frac{u_2^2}{2g}$ corresponding to the impeller velocity it gives the non-dimensional coefficient of pressure $\psi = \frac{H}{u_2^2/2g}$. According to this formula ψ is independent of the density of the flowing medium and of the momentary value of the impeller velocity, for it is

$$H = \frac{\eta_h}{2g} \cdot u_2^2 \text{ (angle constant).}$$

Thus for similar machines with the same Re, identical ψ values are obtained owing to the conformity of η_h and the angles. These non-dimensional coefficients of pressure were fixed by Prandtl as comparative data for fans.

It has been assumed up to this point that the theoretical velocity triangles correspond to the actual conditions, which is the case for a definite flow volume dependent only on the peripheral velocity. If the volume is altered by throttling, however, in similar machines with the same Re the deviations from the theoretical velocity planes are the same when the percentage volume

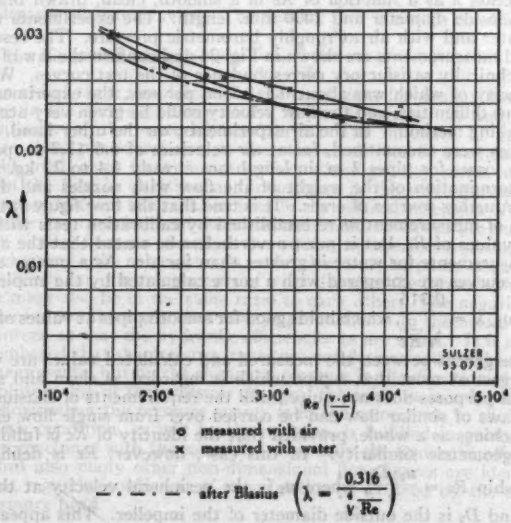


Fig. 24. Coefficients of friction λ in a clean, drawn brass pipe of 25 mm. diameter and 2,000 mm. length.

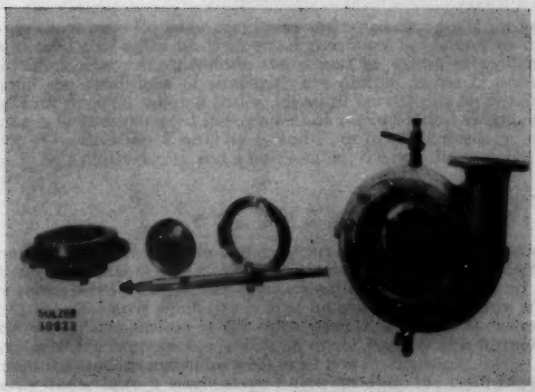
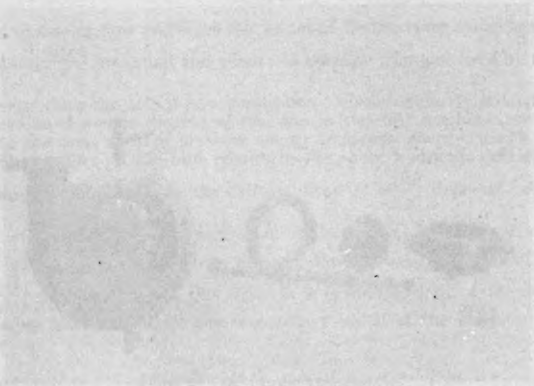
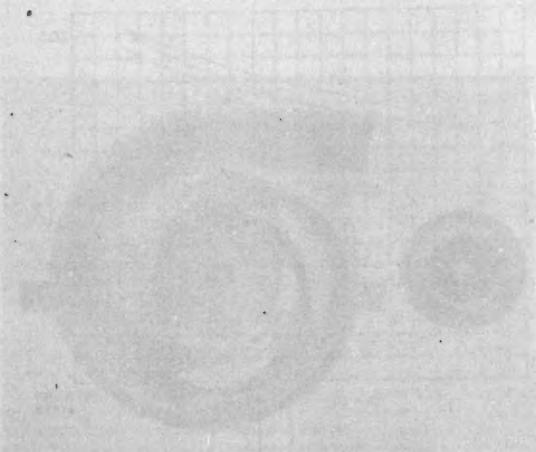


Fig. 25.—Sulzer medium-lift centrifugal pump (delivery Q = 22 lit. per sec.) with the same Reynolds number as the high-pressure centrifugal fan shown in Fig. 26.

Fig. 26.—Sulzer high-pressure centrifugal fan (delivery V = 0.96 cu. m. per sec.) with the same Reynolds number as the medium-lift centrifugal pump shown in Fig. 25.





changes are the same. Equal impact losses and equal changes in the velocities then result, and this again leads to equal resistance heads and to the same hydraulic efficiencies. If, therefore, the ψ and η_h values of two machines which fulfill the conditions of similarity are plotted above the volume as abscissae with unequal volume scales, identical curves can be obtained. If the volume is further rendered independent of the incidental values of u_2 and the section F by dividing V or Q by u_2 and F, or in other words by reducing the problem to a unit velocity and a unit section, the non-dimensional volume coefficient

$$\varphi = \frac{V}{u_2 \cdot F} \text{ or } \frac{Q}{u_2 \cdot F}$$

is obtained. Plotted over φ, the ψ and η_h curves of similar machines with the same Re are thus identical, i.e., the machines furnish like characteristics if these are represented with the non-dimensional magnitudes φ, ψ and η_h. On account of the geometric similarity it is of no essential importance what section is chosen for comparison in the calculation of φ; but it is customary to introduce for F the expression $\pi/4 \cdot D_2^2$, no doubt because this formulation is mainly used for modern machines with axial flow.

To verify these laws a single-stage Sulzer medium-lift centrifugal pump and a geometrically similar high-pressure fan of wooden construction were erected in the machine laboratory of the Winterthur Technical College. The data of the two machines are as follows:—

Pump (Fig. 25):

$Q = 22$ lit./sec.; $H_{man} = 12.85$ meters;
$n = 1{,}480$ Rpm.; $D_2 = 210$ mm.
With water at 18° C. and $\nu = \frac{1.06}{10^6}$ sq. m./sec.,
$Re = 3.2 \cdot 10^6$.

Fan (Fig. 26):

$V_2 = 0.96$ cu. m./sec.; $\Delta p = 435$ mm. W. G.;
$n = 2{,}920$ Rpm.; $D_2 = 580$ mm.
With air at 18° C. and $\nu = \frac{1.5}{10^5}$ sq. m./sec.,
$Re = 3.4 \cdot 10^6$.

They provided the characteristics given in Fig. 27, which may be regarded as satisfactory confirmation of the laws of similitude, particularly when it is considered that the measurements were in part made by six study groups comprising a total of sixty students. The greatest deviation of the pressure coefficients ψ of 7% with closed delivery branch has an adequate explanation in the "pumping" of the fan in this condition at very low delivery quantities, which at once has an unfavorable influence on the conversion of energy. In the centrifugal pump this phenomenon is impossible owing to the incompressibility of water. With regard to the efficiencies it should be pointed out that only the hydraulic and not the mechanical losses are subject to the laws of flow. It was an easy matter in the present case to split up the total losses into those of a hydraulic and those of a mechanical nature, as the impellers of both machines are overhung and can be rapidly removed. The inputs measured without impellers then correspond to the mechanical friction losses. Special attention must be paid in this connection to the pump stuffing-box, the working conditions of which must be the same with and without the impeller. The input of the pump was determined in this case with a pendulum dynamometer, and that of the fan with an Amsler torsion dynamometer. The N_{ad} of the fan is calculated, with due regard for the compressibility of the air at the pressure differences in question, as

$$N_{ad} = \frac{G \cdot v_{mi} \cdot \Delta p}{75} \text{ Hp.}^1$$

[1] v_{mi} = mean specific volume

For instruction purposes it is very valuable to be able to divide up the total efficiency $\eta_{man} = \frac{N_{man}}{Ne}$ or $\eta_{ad} = \frac{N_{ad}}{Ne}$ numerically for once, into the product $\eta_h \, \eta_{mech}$. For the pump, for example, at an $\eta_{man} = 0.764$ the η_h was 0.801 and the $\eta_{mech} = 0.954$. For the purpose of ensuring corresponding values of Re in fan and pump, the latter was run at $n = 1480$ Rpm. only, though normally intended for twice that speed. It then reached an η_{man} of at least 0.78, which was due in part to a favorable relation of N_{man} to the mechanical friction losses and in part to the decrease of the hydraulic friction coefficients (see Fig. 24) with increasing Re.

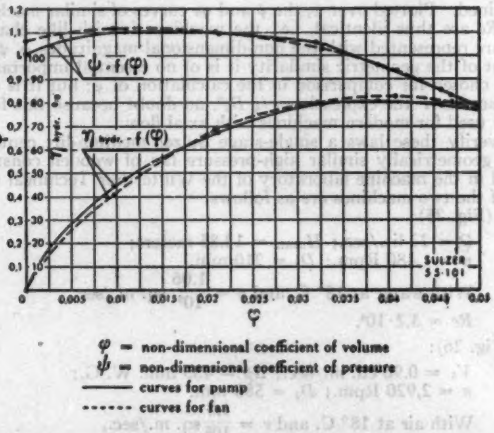


Fig. 27. Characteristics of the medium-lift pump and high-pressure fan shown in figs. 25 and 26.

A typical and instructive example of the application of the laws of similitude in practical engineering is provided by model tests carried out by Sulzer Brothers for the hydraulic storage pumps installed in the Etzel Power Station. Each of the two pumps (Fig. 28), which are the product of a conjoint working enterprise of the Escher Wyss Engineering Works and Sulzers, is built for $Q = 2.77$ cu. m. per sec., $H_{man} = 485.3$ m., $n = 500$ Rpm., $Ne = 20{,}880$ Bhp., 5 stages, impeller diameter $D_2 = 1640$ mm. As very high efficiencies had to be guaranteed and the space conditions set narrow limits to the designer's scope, Sulzer Brothers resolved to construct a single-stage model pump having inlet, impeller, guide vanes and return pipe geometrically similar in form but on the reduced scale of 1 : 2.96. The diameter of the model impeller and the testing speed were first of all fixed so that the peripheral velocity would be the same as in the actual pump, this being necessary with a view to obtaining the same cavitation limit. The Re obtained in this way in the model test with water was only $24 \cdot 10^6$, while in the actual execution the value is about three times as great. As the curves of Fig. 24 show, however, the change in the coefficients of hydraulic friction as a function of Re is slight when the values of Re are great. At all events these coefficients decrease as Re increases, approaching a constant limit. This fact is confirmed by a second series of tests carried out with water, in which Re was only $12 \cdot 10^6$ (see characteristic of model pump, Fig. 29). The higher value of Re is accompanied by a better, if only slightly better efficiency. The efficiencies refer to the power input at

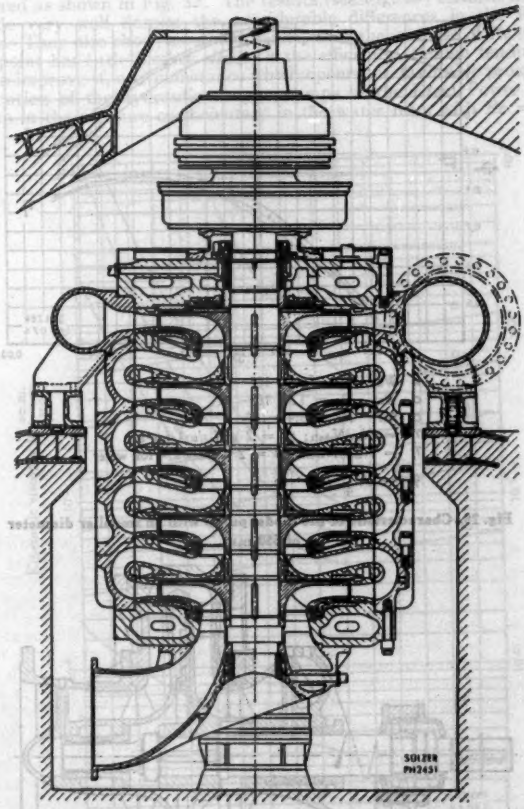
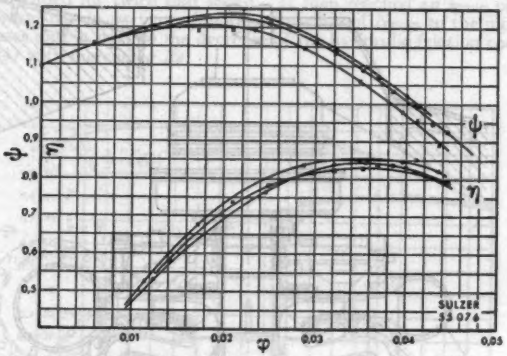


Fig. 28. Section through the 20,500-H. P. hydraulic storage pump built by the Escher-Wyss-Sulzer conjoint working enterprise. The pump delivers 2,700 lit. per sec. against a total head of 491 m. and is intended for the Etzel Power Station.

the shaft and thus include the mechanical losses, so that the better efficiency at higher *Re* can also be understood as a more favorable ratio of the effective performance to the mechanical losses, a point already mentioned in connection with the medium-lift centrifugal pump in tbe machine laboratory of the Winterthur Technical College.

The model of the Etzel storage pump (Fig. 30) was also used for air tests,



× Water $Re = 24 \cdot 10^6$
○ Water $Re = 12 \cdot 10^6$
□ Air $Re = 3.4 \cdot 10^6$
$\eta = N_{tot}/N_{shaft}$; $\psi = 2\,g\,H_{tot}/u_2^2$ for air
$\eta = N_{mano}/N_{shaft}$; $\psi = 2\,g\,H_{mano}/u_2^2$ for water
$\varphi = V/\frac{\pi}{4} D_2^2 \cdot u_2$

Fig. 29. Characteristic of the model pump with an impeller diameter of 550 mm.

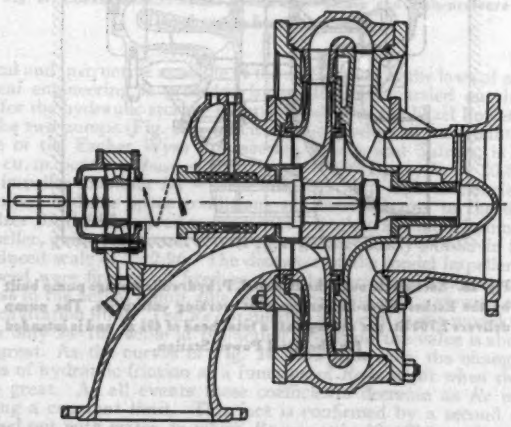
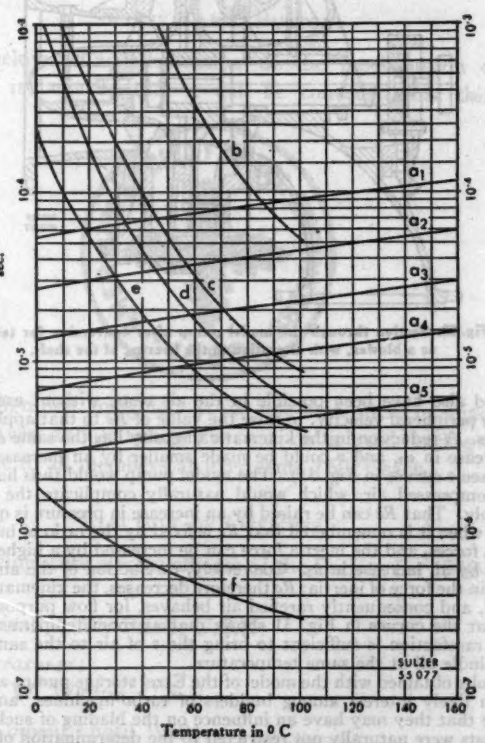


Fig. 30. Section through the model pump with an impeller diameter of 550 mm.

but owing to considerations of impeller strength the value of Re had in this case to be reduced to $3.4 \cdot 10^6$, for the much higher value of the kinematic viscosity ν of air as compared to the ν of water (see ν curves for various materials in Fig. 31) would have required a far greater u_2 with equal Re. In order to reduce the bearing friction losses approximately in proportion with the effective performances in the change-over from water to air, the shaft bearing was altered as shown in Fig. 32. The results (see Fig. 29) confirm the law of similitude very well despite the considerable differences in the Reynolds numbers. They also show, as stated before, that the increase of Re beyond a certain point has little further effect on the efficiency, though tending at all events to improve it, a circumstance which opened up favorable prospects for the execution of the hydraulic storage pumps of the Etzel Power Station. The drop in the pressure coefficients ψ in the water tests with $Re = 24 \cdot 10^6$,



a_1	Air 0.25 atm. abs.	b	Superheated-steam cylinder oil
a_2	Air 0.5 atm. abs.	c	Heavy machine oil
a_3	Air 1 atm. abs.	d	Turbine oil
a_4	Air 2 atm. abs.	e	Spindle oil
a_5	Air 5 atm. abs.	f	Water

Fig. 31. Comparison of kinematic viscosities ν in sq. m. per sec. for air, water and oil.

as compared with the results with lower *Re*, permits of the conclusion that the model pump comes near to the cavitation limit at top speed. In the air tests η_h was also determined separately, as in the tests at the Winterthur Technical College, the astonishingly high value of $\eta_h = 0.841$ being obtained at an $\eta_{ad} = \frac{N_{ad}}{Ne} = 0.831$. (See value measured at $\varphi = 0.03$ in Fig. 29.) The optimum value of η lies higher still.

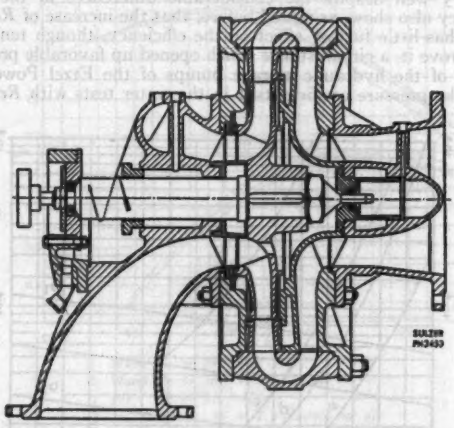


Fig. 32. Section through the model pump after conversion for tests as a blower, with alteration to the bearing of the shaft.

It would also have been possible in the air tests, without exceeding the admissible peripheral velocity, to raise the value of *Re* to that applying in the water tests. A reduction in the kinematic viscosity has the same effect on *Re* as an increase in u_2, and ν could be made smaller by an increase in the air pressure (see ν curves in Fig. 31). The model pump would thus have to work with precompressed air, which would naturally complicate the equipment considerably. That *Re* can be raised by an increase in pressure is quite understandable when it is remembered that *Re* is fixed by the ratio of inertia forces to friction forces, and the inertia force can be increased by a higher density ρ as well as by an increase in u_2. Conversely, rarefaction of the air leads to a reduction in the force of inertia; *Re* therefore decreases, the kinematic viscosity ν is raised, and consequently rarefied air behaves, for flow purposes, like oil. A glance at the curves in Fig. 31 shows that at room temperatures even a moderate rarefaction is sufficient to bring the ν of air to the same value as that of spindle oil at the same temperature.

The results obtained with the model of the Etzel storage pumps are qualified to awaken lively interest among builders of turbo-machines, and it is not impossible that they may have an influence on the blading of such machines. The air tests were naturally not restricted to the determination of the values given here, as the use of air permitted the losses in all flow elements to be measured individually. Since these results with air cannot yet be compared with corresponding results for water, however, they will not be further considered here.

ASSOCIATION NOTES.

Addresses.

It is again urged that each member take steps to ensure that the Secretary is informed of his address.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a

very fine, dignified insignia. It is one-half inch in diameter. The oak leaves and lettering are red on a gold back-ground. It is available to all members at fifty cents (50c) each.

Election of Officers.

The following were elected officers of the Society for 1948-1949.

President:

Rear Admiral R. W. Paine, U.S.N.

Secretary-Treasurer:

Captain F. W. Walton, U.S.N.

Member of Council (until 1950):

Rear Admiral J. W. Fowler, U.S.N.
Captain L. V. Honsinger, U.S.N.
Captain W. N. Mansfield, U.S.N.R.
Mr. M. Douglas Gibson, Jr.

Amendment of By-Laws.

The proposed amendment of the By-Laws to provide for the senior member present of the Council to act as President in the temporary absence of that officer was adopted.

Banquet.

The Annual Banquet of the Society will be held at the Hotel Statler in Washington, on Friday, 7 May, 1948. Announcement of details, with reservation forms, have been distributed.

Membership.

Attention of all non-member officers is especially invited at this time to the advantage and benefit to be attained by membership in this Society. The Society was organized 1888—for the sole purpose of the advancement of Engineering. Its Journal, published quarterly, in February, May, August and November, and entering its 60th year of continuous publication, has attained a foremost rank in its field and is a recognized authority throughout the engineering world. It is used generally by Naval officers and civilians alike as reliable reference book. The Bureau of Ships considers the Journal of such value that it makes it available for the libraries of all Naval vessels and shore activities of any size. The presence of the Journal in your personal library cannot fail to be of great benefit to you, often making available information of great value which can be obtained from no other source.

It is particularly suggested that officers of the Naval Reserve, who are returning to civil life, should arrange for membership prior to detachment from active duty.

Annual dues are $5.00, which are probably lower than those of any other technical Society. There is no initiation fee, nor any extra charge for the Journal. This extremely low cost is possible only because the Society is bona fide operated solely for your benefit without any pecuniary profit to any one.

An application blank generally will be found at the back of each copy of the Journal. However, if none is found, application by letter will be accepted.

It is suggested that Members will confer a real favor on non-members by bringing the Society to their attention.

Membership Additions.

The following have joined the Society since the publication of the November, 1947, Journal:

Naval.

Adams, Charles F., Commander, U.S.N.R.,
Executive Vice President, Raython Mfg. Co.,
Waltham 54, Mass.

Bemis, John Carter, Ensign, U.S.N.R.,
Room 1015, Michelson Laboratory, U.S.N.O.T.S.
Invoken, Calif.

Birkhimer, F. E., Lieut., j.g., U.S.N.R.,
Country Club, Salem, Ohio

Brown, Melvin Wilbur, Lieut., U.S.N.,
5 Hallron Road, Newton Lower Falls 62, Mass.

Buckley, Walter J., Commander, U.S.N.,
c/o Supervising Inspector of Naval Material,
1375 Euclid Ave., Cleveland, Ohio.

Drabic, Anton, Lieut., U.S.N.R.,
Marine Surveyor, 559 Surf St., Chicago, Ill.

FitzGerald, William H., Commander, U.S.N., Ret.
1741 K St., N.W., Washington, D. C.

Frawley, Joseph E., Lieut. Commander, U.S.N.R.,
Associate Engineer, Western Electric Co.,
Mail 115 Willow St., Brooklyn 2, N. Y.

Haffenden, C. R., Captain, U.S.N.R.,
International Diesel Electric Co., Inc., 13-02 44th Ave.,
Long Island City 1, N. Y.

Hanson, H. C. Machinist, U.S.N., BOQ, U. S. Naval Station,
Tongue Point Astoria, Oregon.

Harvey, James L., Chf. Mach., U.S.N.

Hicks, George F., Captain, U.S.C.G.,
Coast Guard Headquarters, Washington, D. C.

Howard, W. E., Jr., Captain, U.S.N.,
Code 140, Norfolk Naval Shipyard, Portsmouth, Va.

Ingwersen, Martin L., Lieut., j.g., U.S.N.R.,
Great Lakes Engineering Works, Ashtabula, Ohio,
Mail RD No. 3, Madison, Ohio.

Jakubowski, Thaddeus J., Lieut., U.S.N.

Krause, Ralph A., Commander, U.S.N.R., Asst. to President,
Raytheon Mfg. Co., Waltham 54, Mass.

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Shriver, William A., Lieut., U.S.N.

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Principal Owner Luzon Stevedoring Co., P. O. Box 582,
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Kuhn, Raymond W., Jeannette, Pa.

McCandless, Lyon H., President, American-Indiana, Inc., 420 Madison Ave., New York 17, N. Y.

Norgaard, Loring C., Nav. Arch. Partner, Norgaard & Clark, Pier 5, Outer End., San Francisco, Calif.

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TRANSFERRED.

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Associate to Civil: Wilson, R. M., Westinghouse Electric Corp., 1625 K St., Washington, D. C.

DEATHS.

It is with much regret that the following deaths among our membership during the past year are announced:

NAVAL.

Andre', Rufus Luke
Barry, Ralph E.
Kellogg, Edwin J.
Penn, Albert M.
Shea, H. E.
Wylie, Harold

CIVIL.

Collins, W. H.
Chadwick, George W.
Ferrier, George B.
Jerguson, Phillip A.
Kothny, G. L. E.
Long, Richard R.

ASSOCIATE.

Fay, H. J. W.
Oldham, P. F.
Sadtler, Edwin B.